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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

SEMIANNUAL TECHNICAL PROGRESS REPORT FOR PERIOD: JULY 1 – DECEMBER 31, 1982

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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

Report No. 83ASE308SA3

Semiannual Technical Progress Report

Period Covered:
July 1 - December 31, 1982

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INTRODUCTION

In March, 1978, a Stirling engine development contract, sponsored by the Department of Energy and administered by NASA/Lewis Research Center, was awarded to Mechanical Technology Incorporated (MTI) for the purpose of developing an automotive Stirling engine and transferring Stirling engine technology to the United States. The program team consists of MTI as prime contractor, contributing their program management, development, and technology-transfer expertise; United Stirling of Sweden (USAB) as major subcontractor for Stirling engine development; and AM General (AMG) as the major subcontractor for engine/vehicle integration.

Most Stirling engine technology previously resided outside of the United States, and was demonstrated for stationary and marine applications; therefore, the Automotive Stirling Engine (ASE) Development Program was directed at the establishment and demonstration of a base of Stirling engine technology for automotive application by September, 1984. The high-efficiency, multifuel capability, low-emissions, and low-noise potential of the Stirling engine make it a prime candidate for an alternative automotive propulsion system.

ASE Program logic called for the design of a Reference engine to serve as a focal point for all component, subsystem, and system development within the program. The Reference Engine System Design (RESD), defined as the best-engine design generated at any given time within the program that will provide the highest possible fuel economy and meet or exceed all other program objectives, utilizes all new technologies that are reasonably expected to be developed by 1984, and that are judged to provide significant improvements relative to the risk and cost of their development.

The Mod I and Mod II engines are experimental versions of the RESD. The Mod I was the first engine design that used existing technologies embodied in USAB's P-40 and P-75 engines. The Mod II (slated to be designed based on the RESD, the Mod I, and component development improvements) is an engine design directed toward meeting the final ASE Program objectives.

In March, 1981, the RESD was updated to predict a combined mileage of 41.1 mpg on unleaded gasoline (55% above projected internal-combustion-engine mileage) for a 1984 X-body vehicle with a curb weight of 2870 pounds; however, because of Government funding cutbacks, the Mod II design and associated development efforts were never started, making the Mod I the only experimental engine in the ASE Program.

Since the Mod II would not be designed, it was reasoned that the Mod I could be used to develop and demonstrate RESD technology through a series of design upgrades, i.e., it would be more cost-effective to use existing Mod I hardware than to design and build an entirely new set of Mod II engine hardware; as a result, the "Proof-of-Concept" logic evolved.

Mod I engine hardware is to be used to "prove" the designs and technologies embodied in the RESD. In order to prove the RESD concepts, the necessity of conducting two upgraded designs of the Mod I was recognized. These two upgraded versions were identified as the Mod I-A and Mod I-B. Inherent limitations were also recognized in the proof-of-concept since Mod I hardware was larger, and of a fundamentally different design than RESD and Mod II hardware.

During the past six months, the ASE Program directed its resources primarily toward the characterization of Mod I engines, completion of the design of the Mod I-A engine systems, and component development/design substantiation. Effort was also directed toward updating the RESD design in order to reduce its cost and improve its reliability.

The four current ASE Mod I engines have accumulated a total of 1700 test hours. The U.S.A.-manufactured/assembled Mod I accumulated 217 hours in just four months, while the engine installed in the AMC Lerma vehicle has accomplished over 200 starts and over 120 hours of transient operation.

I. SUMMARY

Since the inception of the ASE Program in 1978, 13 Quarterly Technical Progress Reports have been issued under NASA Contract No. DEN3-32, "Automotive Stirling Engine Development Program;" however, reporting was changed to a semiannual format in July, 1981. This report, the third Semiannual Technical Progress Report issued under the contract, and covering the period of July 1 to December 31, 1982, includes technical progress only. Although the program has been modified to a proof-of-concept format, the objectives described below still apply to the RESD. The upgraded versions of the Mod I engines are not, however, required to demonstrate all these hardware objectives.

Overall Program Objectives

The overall objective of the ASE Program is to develop an automotive Stirling Engine System (SES) by September, 1984 which, when installed in a late-model production vehicle, will:

- demonstrate at least a 30% improvement in combined metro/highway fuel economy over that of a comparable spark-ignition, engine-powered production vehicle, based on EPA test procedures*; and,

- show the potential for emissions levels less than: $NO_x = 0.4$ g/mi, HC = 0.41 g/mi, CO = 3.4 g/mi, and a total particulate level of 0.2 g/mi after 50,000 miles.

In addition to the above objectives, which are to be demonstrated quantitatively, the following system design objectives are also considered:

- ability to use a broad range of liquid fuels from many sources, including coal and shale oil;

- reliability and life comparable to current-market powertrains,

- a competitive initial cost and life-cycle cost comparable to a conventionally powered automotive vehicle;

- acceleration suitable for safety and consumer considerations, and,

- noise/safety characteristics that meet the currently legislated or projected Federal Standards for 1984.

Major Task Descriptions

The overall objectives of the major program tasks are described below as modified for the proof-of-concept program:

Task 1 - Reference Engine - This task, intended to guide component, subsystem, and engine system development, involves the establishment and continual updating of an RESD, which will be the best engine design that can be generated at any given time, and that can provide the highest possible fuel economy while meeting or exceeding other final program objectives. The engine will be designed for the requirements of a projected reference vehicle that will be representative of the class of vehicles for which it might first be produced, and it will utilize all new technology (expected to be developed by 1984) that is judged to provide significant improvement relative to the risk and cost of its development.

Task 2 - Component/Technology Development - Guided by RESD activities, this task will be conducted in support of various Stirling engine systems, and will include conceptual and detailed design/analyses, hardware fabrication and assembly, and component/subsystem testing in laboratory test rigs. When an adequate performance level has been demonstrated, the component and/or subsystem design will be configured for in-engine testing and evaluation in an appropriate engine dynamometer/vehicle test installation. The component development tasks, directed at advancing engine technology in terms of durability/reliability, performance, cost, and manufacturability, will

*Automotive Stirling and spark-ignition engine systems will be installed in identical model vehicles that will give the same overall vehicle driveability and performance.

include work in the areas of combustion, heat exchangers, materials, seals, engine drivetrain, controls, and auxiliaries.

Task 3 - Technology Familiarization (Baseline Engine) - The existing USAB P-40 Stirling engine will be used as a baseline for familiarization, as a test bed for component/subsystem performance improvement, to evaluate current engine operating conditions/component characteristics, and to define problems associated with vehicle installation. Three P-40 engines will be built and delivered to the United States' team members; one will be installed in a 1979 AMC Spirit. A fourth P-40 engine will be built and installed in a 1977 Opel sedan for testing in Sweden. The baseline P-40 engines will be tested in dynamometer test cells and in the automobiles. Test facilities will be planned and constructed at MTI to accommodate the engine test program and required technology development.

Task 4 - Mod I Engine - A first generation automotive Stirling engine (Mod I) will be developed using USAB P-40 and P-75 engine technology as an initial baseline upon which improvements will be made. The prime objective will be to increase power density and overall engine performance. The Mod I engine will also represent an early experimental version of the RESD, but will be limited by the technology that can be confirmed in the time available. The Mod I need not achieve any specific fuel economy improvements. It will be utilized to verify concepts incorporated in the RESD, and to serve as a stepping stone toward the Mod II engine, thus providing an early indication of the potential to meet the final ASE Program objectives.

Three engines will be manufactured in Sweden and tested in dynamometer test cells to establish their performance, durability, and reliability. Continued testing and development may be necessary to meet preliminary design performance predictions. One additional Mod I engine will be manufactured, assembled, and tested in the United States.

A production vehicle will be procured and modified to accept one of the above engines for installation. Tests will be conducted under various steady-state, transient, and environmental conditions to establish engine-related driveability, fuel economy, noise, emissions, and durability/reliability.

The Mod I engine will be upgraded through design improvements to provide a "proof-of-concept" demonstration of selected advanced components defined for the RESD. Two upgraded versions of the Mod I (Mod I-A and Mod I-B) will be conducted.

Task 5 - Mod II Engine - Deleted from the program.

Task 6 - Prototype Study - Deleted from the program.

Task 7 - Computer Program Development - Analytical tools will be developed that are required to simulate and predict engine performance. This effort will include the development of a computer program specifically tailored to predict SES steady-state cyclical performance over the complete range of engine operations. Using data from component, subsystem, and engine system test activities, the program will be continuously improved and verified throughout the course of the program.

Task 8 - Technical Assistance - Technical assistance will be provided to the Government as requested.

Task 9 - Program Management - Work under this task will provide total program control, administration, and management, including reports, schedules, financial activities, test plans, meetings, reviews, and technology transfer.

Program Schedule

A current schedule of the major milestones for the ASE Program is presented on the following page.

MILESTONES

FY 1981	FY 1982	FY 1983	FY 1984	FY 1985
Complete Steady-State Characterization of Mod I Build 1				
	Complete Mod I Transient Evaluation			
		Complete Steady-State Characterization of First Upgrade (Mod I-A)		
			Complete Endurance Test of Mod I-A	
			Complete Mod I-A Transient Evaluation	
				Technology Readiness Assessment
				Reference Engine Design Update

Program Status and Plans

A summary of the accomplishments achieved in the ASE Program during this semiannual report period are presented in the following sections.

TECHNOLOGY DEVELOPMENT

Development of a dual-orifice Delavan fuel nozzle continues. Tests in the Combustion Performance Rig were completed, showing acceptable emissions levels and temperature profiles in an exhaust gas recirculation (EGR) combustor. Hardware has been fabricated and is being assembled for full engine testing to evaluate transient characteristics of the nozzle.

Development of two alternate regenerator materials continues. Mod I engine-size regenerators manufactured from Metex were installed in engine No. 10 for evaluation. Performance is expected to be slightly reduced; however, manufactured cost of Metex is ~20% of the current design (uses sintered mesh screen). The second alternate material is a silicon carbide (SiC) foam commercially called Duocel. Early rig test data show efficiencies equal to or better than sintered screen mesh at ~10% of the cost of the current design. Material samples are currently undergoing durability testing in a rig; engine testing is scheduled.

Development of a ceramic preheater has been initiated. Rig tests of have been initiated on two test samples received from Coors to determine heat-transfer and pressure-drop characteristics for various airflow rates. Durability tests, which will simulate cold-start transients, will be conducted to establish the affect of rapid increases in temperature on the life of the samples.

As a result of an extensive review of main seal reliability/durability, new procedures have been instituted throughout the program. An assembly procedure has been put into effect to ensure that seals are installed in a wet condition. Specific dimensional data and weight will be recorded before and after seals are installed in an engine.

Three P-40 engines have been assigned for test to provide statistical data on the life of Pumping Leningrader (PL) seals. These engines have been modified to exactly duplicate the Mod I PL seal except for stroke. Each engine will be tested according to a prescribed duty cycle to properly simulate cold starts and high-pressure/velocity environments. When each set of seals completes 500 hours of testing, they will be replaced by a new set.

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As shown in Figure 1-1, P-40 engine No. 4 is currently testing a second set of PL seals after a first set completed the required 500-hour test. P-40 engine No. 9 has a set of seals under test, while engine No. 5 has yet to begin testing.

The Lightweight Reduced Friction Drive, which utilizes lightweight crankshafts and rolling-element bearings, has been assembled, and its initial functional checkout has begun. This bearing design permits full initialization of the low-rpm/high-torque capability of the Stirling engine, and provides a 2-kW (2.7-hp) reduction in parasitic friction. This drive configuration will also reduce cold-start penalty because of reduced viscous friction at start-up.

MOD I ENGINE TEST PROGRAM

Four ASE Mod I engines are currently undergoing testing. As shown in Figure 1-2, a total of 1700 test hours have been accumulated to date. Engines No. 1, 2, and 3 were procured and assembled at USAB, whereas engine No. 10 was manufactured in the U.S. and assembled at MTI. The number 10 was assigned to the MTI-assembled engine to identify it from the European-manufactured/USAB-assembled engines (Nos. 1, 2, and 3). The manufacture, assembly, and test of engine No. 10 marked a distinct demonstration of the level of Stirling-engine technology transfer to the U.S. During the procurement phase for engine No. 10, more than 30 U.S. vendors participated.

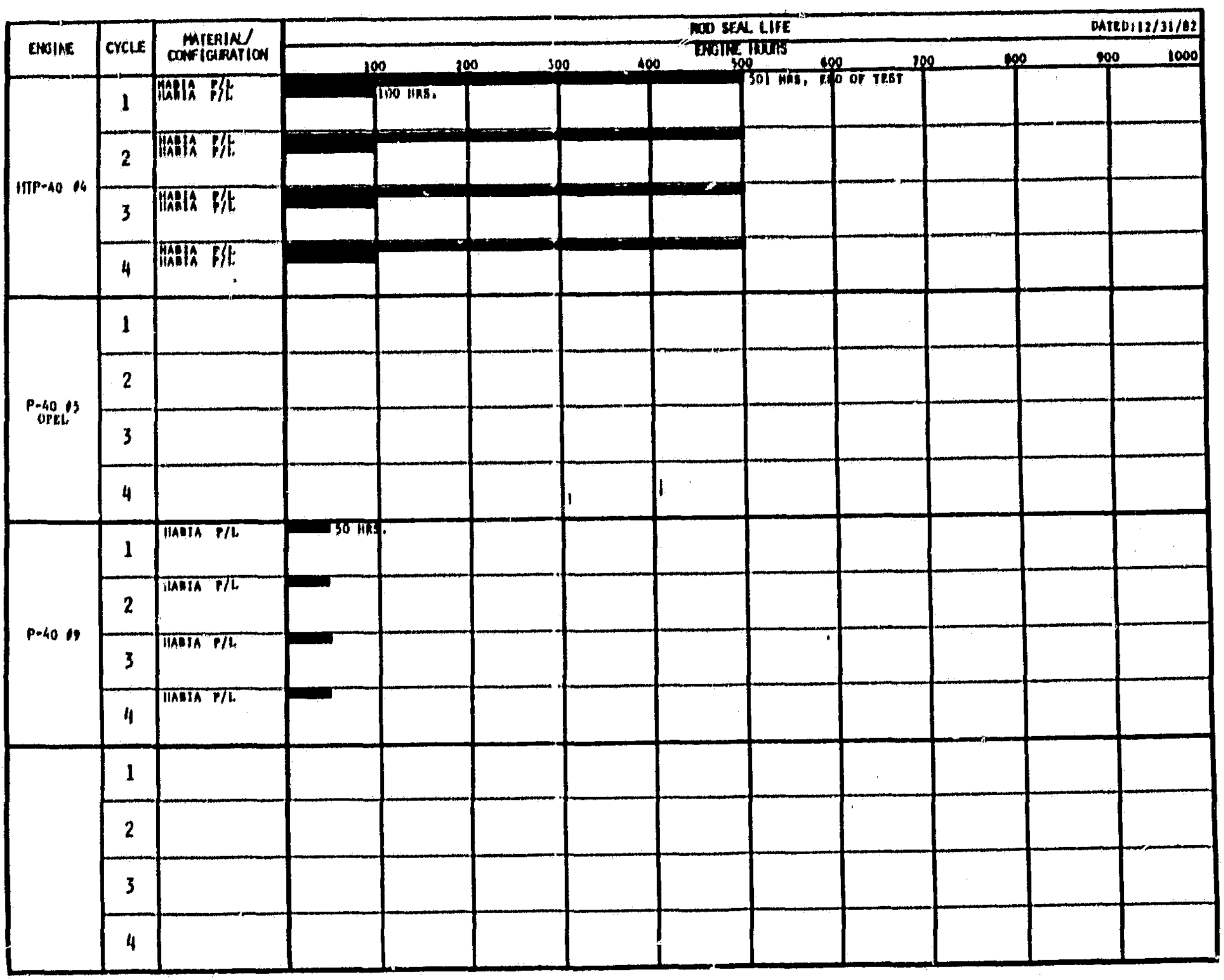

Figure 1-1 Rod Seal Life

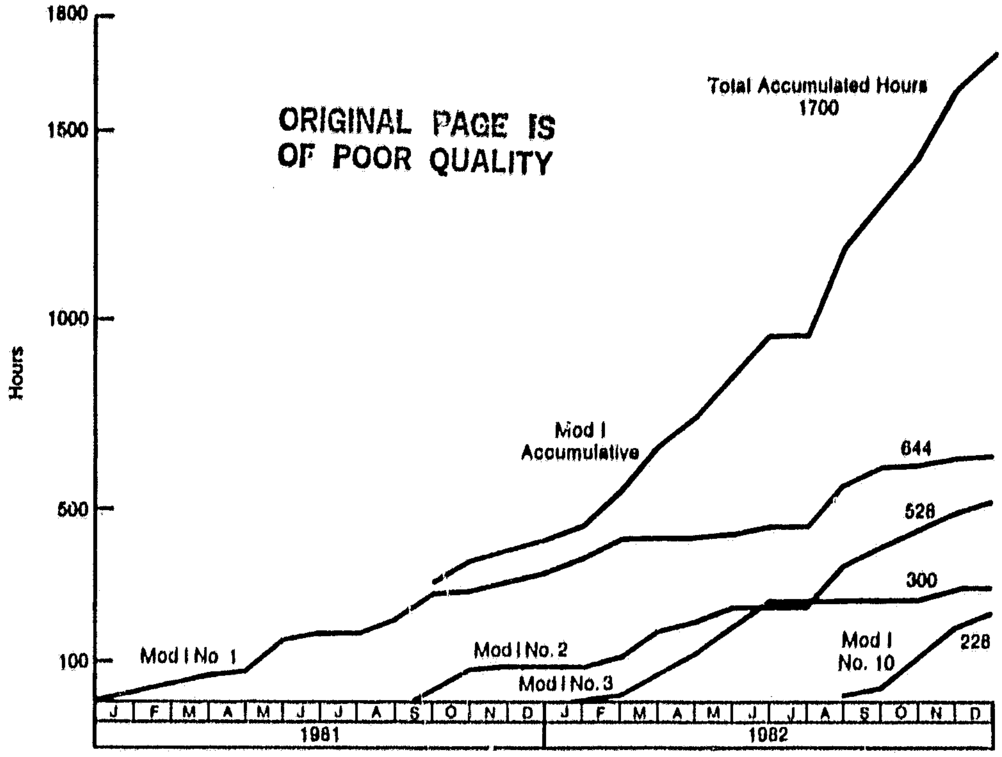

Figure 1-2 Mod I Engine Operating History

After an acceptance test with a new digital control, engine No. 1 was shipped to AM General in early February for installation in a Lerma vehicle designed to serve as a transient test bed (TTB). This installation, completed by the end of May when initial testing began, helped to achieve a major program milestone by allowing the transient characteristics of the Mod I engine to be evaluated and accomplished on schedule by the end of September, 1982. As of the end of December, 1982, engine No. 1 had accumulated a total of 644 test hours.

Three separate Constant Volume Sampling (CVS) tests were conducted on the TTB. Results showed repeatability and fuel economy consistent with predictions. HC and CO emissions levels met program goals; NO_x emission levels also compared favorably with predicted levels.

The EGR system used in the current Mod I gives NO_x levels above the program goal of 0.4 g/mi. With further combustion system development in the Mod I-A engine, this goal will be met by virtue of the higher mileage predicted for the RESD as compared to the Mod I mileage.

Engine No. 2 has remained at USAB, where it completed a series of development tests and accumulated a total of 528 test hours. This engine was assembled initially as a Basic Stirling Engine* (BSE), thus providing the capability of measuring engine power and efficiency without the interference of auxiliary system losses. Engine No. 2 was tested to evaluate an EGR combustor system, and then reconfigured as a Stirling Engine System** (SES). Data obtained from this testing provide back-to-back comparisons of BSE/SES power and efficiency losses.

*complete engine less auxiliaries
**complete engine with auxiliaries

Engine No. 3, currently being reassembled at USAB to begin a 2000-hour endurance test program, has accumulated a total of 300 test hours to date.

Engine No. 10 began testing at MTI in August, 1982, successfully passing acceptance testing in early November. Power and efficiency were above the average of engines No. 1, 2, and 3. To date, engine No. 10 has accumulated 228 test hours. Important test data have been obtained from this engine on back-to-back comparisons of combustion gas recirculation (CGR) and EGR combustor systems, improved air/fuel system components, and development of a Digital Engine Control.

MOD I ENGINE CHARACTERIZATION/ANALYSIS

Figure 1-3 shows power characteristics for the engines tested to date in the Mod I development program; for the sake of clarity, only the 15- and 5-MPa mean charge pressure levels are included. The 15-MPa performance curve represents the maximum available power level, while the 5-MPa data provides performance indications at average operating conditions in a vehicle system.

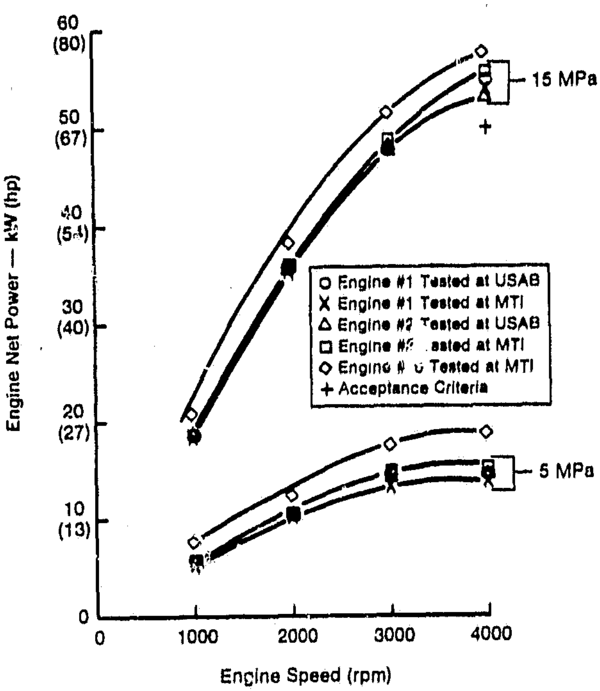

Figure 1-3 Mod I Stirling Engine Net Power

Total engine-to-engine power variation was less than 2 kW (less than 2.7 hp) at all power levels; the maximum power level demonstrated by all engines also exceeded the minimum level established as final acceptance criteria. Figure 1-4 shows the efficiency levels attained during Mod I engine testing. In comparison to power variations, the variation in efficiency levels from engine to engine was greater.

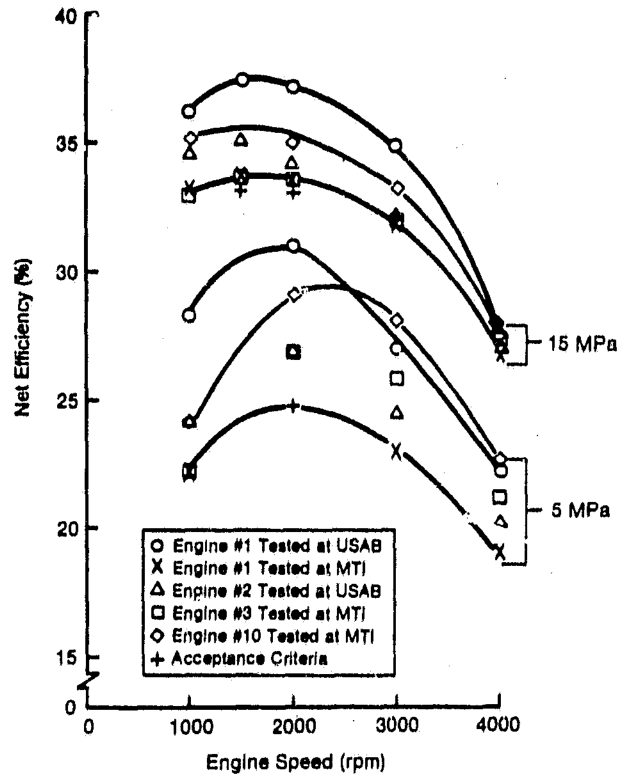

Figure 1-4 Mod I Stirling Engine Net Efficiency

Engine No. 1 showed the most significant deviation in efficiency level during final acceptance testing at USAB. Although the reason for this variation is not completely understood at this time, three contributing factors serve to qualitatively explain the higher efficiency level of engine No. 1:

- This engine ran with the initial version of the CGR combustor, for which higher temperature spreads were measured on the heater head tubes and the working gas temperature. Although measured average heater head temperature level was maintained equal to that used during other engine testing, the actual effective working gas temperature may have been higher than

indicated because of the incomplete temperature coverage on the heater head.

- Following steady-state acceptance testing, transient tests indicated the desirability of modifications to achieve more rapid changes in engine power during deceleration. This effect was achieved by enlarging the internal lines leading to the working gas removal system, thus increasing dead volume in the engine cold space, and resulting in a lower steady-state efficiency level. (All other engines have since incorporated the enlarged internal lines, as did engine No. 1 for its testing at MTI and for the Lerma TTB installation.)

- During final acceptance testing, engine performance may have been affected by warped spacer rings above the regenerators, causing changes in hot-space dead volume, and altering flow distribution through the regenerator.

Engine No. 10, which was configured with a CGR combustor system, showed efficiency levels higher than engines No. 2 and 3, but not to the levels tested earlier in engine No. 1 which, when tested at MTI, utilized an EGR combustor, enlarged internal lines, and nonwarped spacer rings. It should be noted that performance of the engine in this configuration agrees very well with that demonstrated by engines 2 and 3. Furthermore, efficiency levels of all engines met or exceeded those established as acceptance criteria.

During initial testing of engine No. 1 with a CGR combustor (Figure 1-5), high heater head temperature spreads were encountered. Due to the rigid construction of the CGR combustor, warping of combustor mounting and sealing hardware was also encountered during rapid thermal transients of engine starts; therefore, a conclusion was reached that further CGR development was required, and EGR combustors would serve as interim combustors for the Mod I engine. As shown in Figure 1-5, heater head temperature spread is drastically reduced with the EGR combustor.

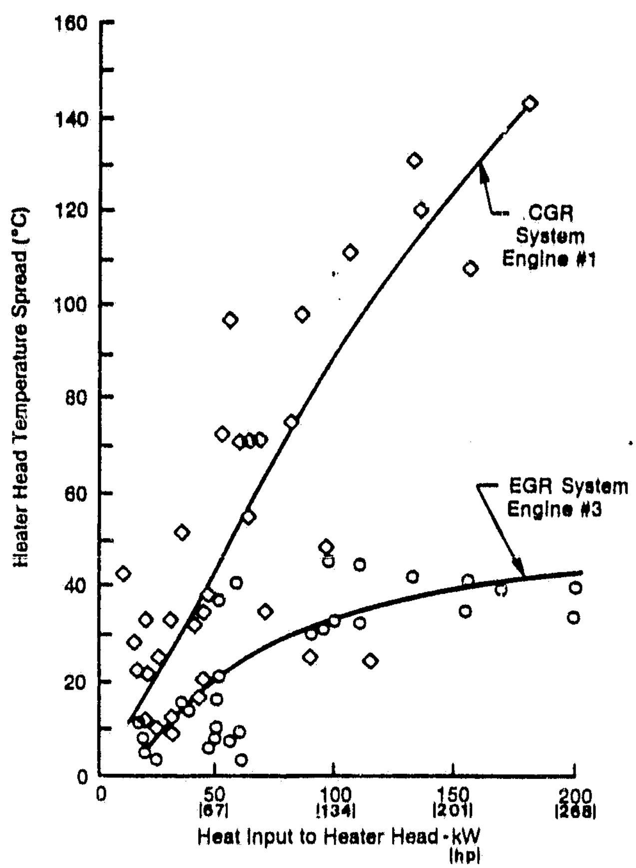

Figure 1-5 EGR/CGR Heater Head Temperature Spread

Testing of an EGR system for the Mod I engine was accomplished during this semi-annual report period using two engines to establish and verify an optimum EGR schedule for providing acceptable emissicns control and engine performance. Performance impact of the selected EGR schedule is shown in Figures 1-6 and 1-7.

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Figure 1-6 illustrates the degradation of engine power with increasing amounts of EGR (the selected EGR schedule is superimposed on this curve, showing a negligible affect on power level). Similarly, Figure 1-7 shows engine net efficiency degradation with increasing EGR, including the selected EGR schedule. EGR impact at the maximum efficiency point is small (0.3 percentage points) and, at the average operating point (AOP), shows an efficiency degradation of 0.5 percentage points. A comparison of EGR and CGR combustor emissions characteristics showed a decrease in CO emissions, and an increase in NO_x emissions; HC emissions were negligible with both combustors.

As illustrated on Figure 1-8, the EGR system provides improved CO emissions characteristics at the higher fuel flow rates. The NO_x comparison (Figure 1-9) shows that emissions are higher with the EGR combustor than with the CGR combustor.

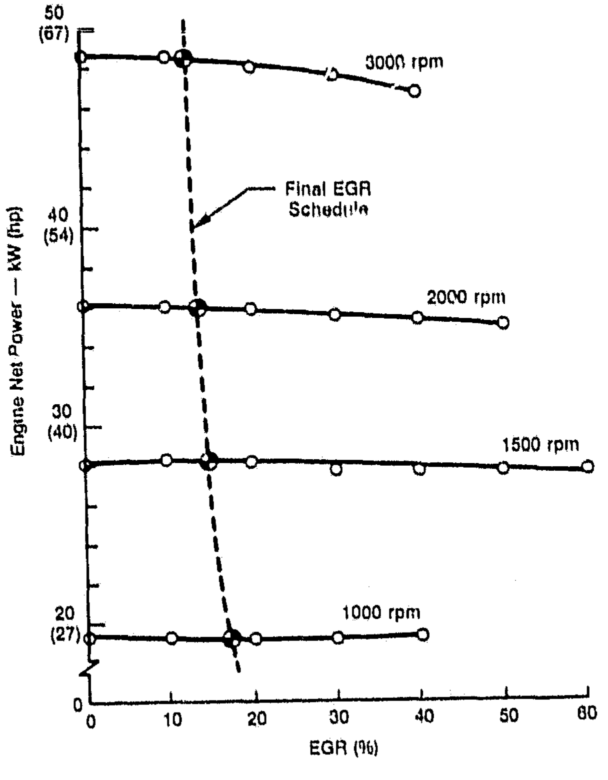

Figure 1-6 Effect of EGR on Power (15 MPa)

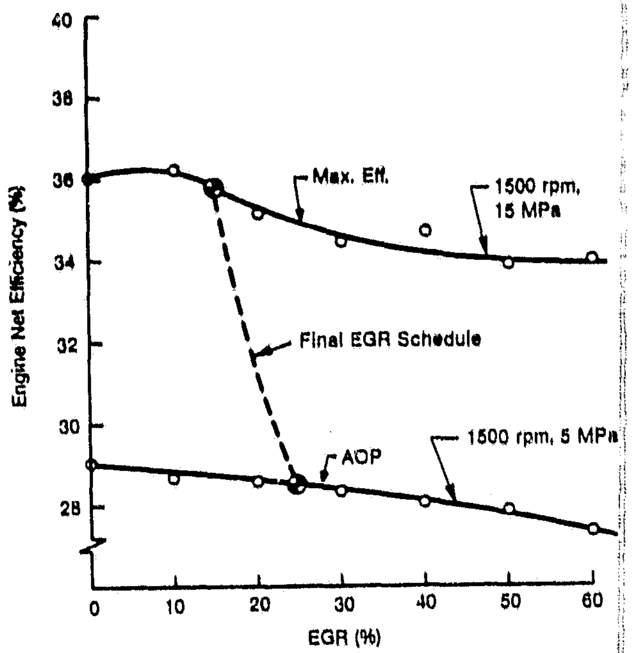

Figure 1-7 Effect of EGR on Efficiency

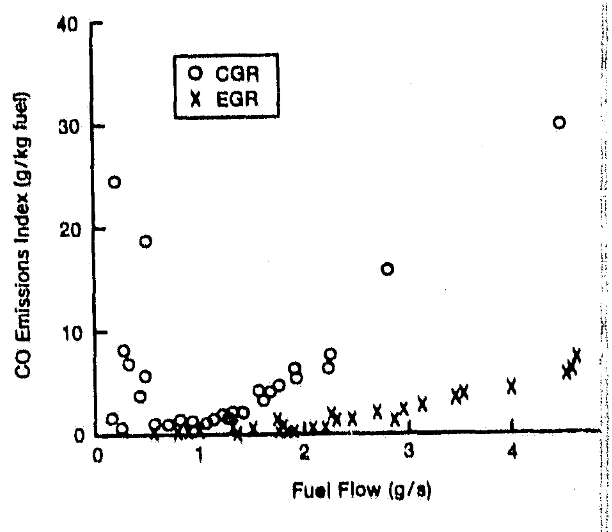

Figure 1-8 CO Emissions

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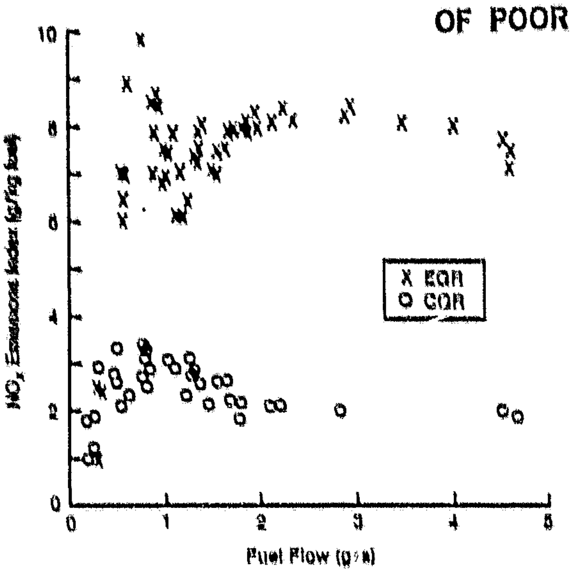

Figure 1-9 NO_x Emissions

MOD I TRANSIENT TEST BED FUEL ECONOMY

Mod I Engine No. 1 is dedicated to the development of transient systems within the ASE Program. To provide this capability, the engine has been installed in an AMG Lerma TTB for optimization of engine/vehicle system characteristics and development of the various engine control systems. Performance improvements realized through this development process can be measured by improvements in vehicle fuel economy; hence, an initial TTB mileage evaluation was established to provide a baseline for testing, and extensive documentation was acquired to enable assessment of any system changes. First, engine No. 1 was fully characterized in MTI's test facility prior to installation in the Lerma TTB. Emissions and standard engine performance data were recorded and utilized to provide baseline engine maps for computer code assessment of vehicle mileage and comparison to any engine modifications accomplished during development. (Engine characterization will be repeated in the event that significant hardware modifications are incorporated.)

Following characterization and installation, baseline CVS tests were accomplished, and analytical mileage projections were established utilizing MTI's vehicle simulation code. Three separate tests, each consisting of a cold-start urban and highway cycle, were conducted. Test results to date, along with projections, are shown in Table 1-1.

TABLE 1-1

LERMA VEHICLE TEST DATA

Description	Urban				Highway	Combined
	HC g/mi	CO g/mi	NO_x g/mi	Mileage mpg	mpg	mgp
9/21 Test	0.23	3.4	0.96	19.9	31.7	23.9
9/24 Test	0.29	3.3	0.90	18.8	32.1	23.2
9/25 Test	0.25	3.2	0.84	19.2	32.4	23.5
Standard Deviation	0.025	0.082	0.049	0.45	0.29	0.29
Mean	0.26	3.3	0.90	19.3	32.1	23.5
Projections			0.84	19.1	30.0	22.8

11.1 hp @ 50 mph Dynamometer Power Setting
3750 lb Inertia Weight Setting
2.73 Axle Ratio

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Excellent agreement with projections, including both mileage and NO_x emissions levels, has been achieved. The data repeatability for the three runs was excellent, with a standard deviation in mileage of 1-2%.

The progress of the ASE Program, as measured by this baseline testing, is shown in Figure 1-10. A direct comparison of as-recorded fuel economy between Mod I data and previous P-40 vehicle data are invalid due to the differences in test weights and power-to-weight ratios for the three vehicles; accordingly, the P-40-powered vehicles have been analytically adjusted in Figure 1-10 to show performance that would result at the same power-to-weight ratios and test weights. As adjusted, Mod I baseline testing represents a 50 and 26% mileage improvement over the P-40 Opel and Spirit vehicles, respectively.

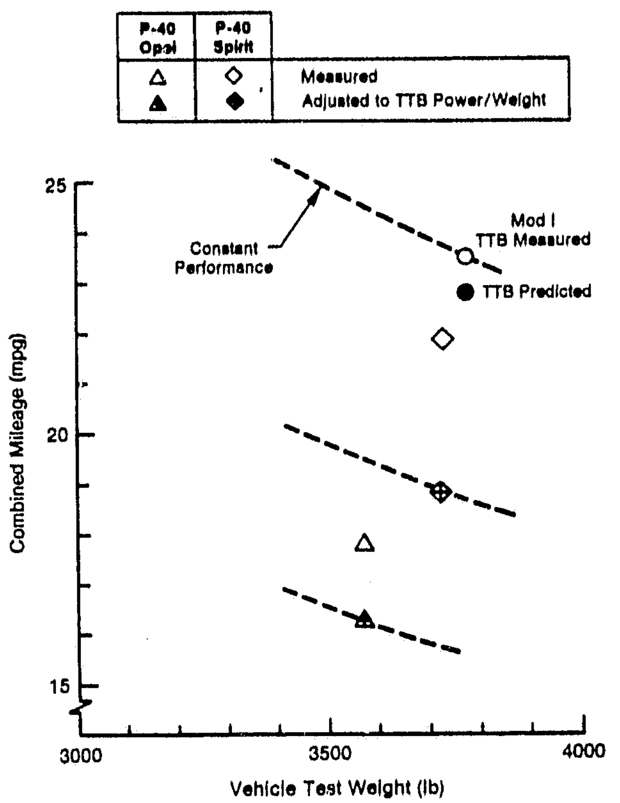

Figure 1-10 Vehicle Fuel Economy Comparison

MOD I-A ENGINE PERFORMANCE

The Mod I-A engine design is aimed at providing improved performance levels relative to the Mod I, and continued proof-of-concept demonstration of the RESD, a projected advanced ASE design.

One goal of the RESD, and therefore the Mod I-A, is the use of low-cost materials void of strategic elements. Design of the Mod I-A using nonstrategic materials in the heater head was a major step toward achieving this goal. The Mod I-A cylinder and regenerator housings would be cast from XF-818 (an iron-based material) rather than from the cobalt-based alloy HS-31 used in the current Mod I design. Tube material was changed from the cobalt-based alloy N155 Multimet to CG-27. Table 1-2 compares the dramatic reduction in the use of strategic materials in the Mod I-A design to that of the current Mod I.

TABLE 1-2

REDUCTION IN STRATEGIC ELEMENTS IN MOD I-A DESIGN

Strategic Element	Mod I	Mod I-A
Cobalt	22.4 lb	—
Chromium	36.4 lb	23.2 lb

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In addition to the normal design goal of improved cycle efficiency, the Mod I-A design effort is addressing the energy consumption losses determined by Mod I Lerma system testing. Table 1-3 summarizes the analytically projected improvements in the Mod I-A engine, and compares them to current analytical modelling of the Mod I engine.

TABLE 1-3

PERFORMANCE PROJECTION

MOD I-A STIRLING ENGINE SYSTEM

Operating Condition	Parameter	Mod I-A	Δ from Mod I
15 MPa 4000 rpm (Maximum Power)	Net Power Net Efficiency	60.26 kW 29.9%	+10.2 kW +2.7 pt
15 MPa 1500 rpm (Maximum Efficiency)	Net Efficiency	40.76%	+3.07 pt
5 MPa 2000 rpm (Average Operating Point)	Net Power Net Efficiency	12.57 kW 33.04%	+2.43 kW +4.95 pt
	Specific Weight kg/kW lb/hp	 4.4 7.3	 -24%

Several design changes (see Table 1-4) contributing to these maximum power and efficiency level improvements include:

- increased set temperature to provide improved cycle efficiency;
- redesign of heater head and regenerator to achieve improved castability and more optimum part-power performance with minimal affect on maximum engine power output;
- the incorporation of a rolling-element bearing drive unit and improved seal design to decrease friction losses; and,
- incorporation of a new fuel nozzle system to eliminate air-atomizer compressor and associated power loss.

TABLE 1-4

MAJOR IMPROVEMENT AREAS

MOD I-A STIRLING ENGINE SYSTEM

Change	ΔMaximum Net Power	ΔMaximum Net η
Increased Set Temperature (720° → 820°C)	+7	+1.0% pt
Part Power Optimization	-2	+1.0% pt
Decreased Friction Losses	+1.2	+0.7% pt
Elimination of Atomizer Air Compressor	+0.5	+0.3% pt

In addition to these steady-state performance improvements, a reduction in cold-start penalty is projected as a result of the lighter-weight, redesigned heater head and External Heat System. A 17% reduction in stored heat (translates to a 0.5 mpg urban mileage improvement) is expected to be achieved with the Mod I-A design. The overall projection for the Mod I-A engine is that combined fuel economy will improve ~3 mpg, or 13% relative to the current Mod I system.

Design of the Mod I-A has been completed, and hardware procured. Currently, two engines are undergoing assembly for demonstration testing in early 1983.

REFERENCE ENGINE SYSTEM DESIGN (RESD)

As stated earlier, the RESD is defined as the best engine design that can be generated at any given time that can provide the highest possible fuel economy while meeting or exceeding other final ASE Program objectives. One of these objectives is a competitive manufacturing cost. A value-engineering design effort was initiated on the RESD that emphasized the need for a manufacturing cost reduction. All proposed design changes were evaluated against the baseline cost established from the study. The following material substitutions reduced the cost of specific RESD components:

- heater head tube material changed from Inconel 625 to iron-base CG-27;
- gas cooler tube material changed from stainless steel to phosphate-coated carbon steel; and,
- preheater matrix material changed from Sandvik 253MA to Armco 12SR or 18SR.

Further, several design concepts were incorporated to reduce the RESD manufacturing cost:

- one-piece, equal-angle, cast-iron V-block;
- perforated plate-gas cooler;

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- one-piece piston dome and rod assembly; and,
- balance shaft elimination.

Table 1-5 summarizes the manufacturing cost reductions associated with each of these concepts.

Table 1-5

MANUFACTURING COST REDUCTIONS

Material Substitutions		Design Changes		Total Cost Reduction
Heater Head Tube	$24	One-Piece V-Block	$25	
Gas Cooler	202	Gas Cooler Configuration	21	
Preheater Matrix	24	Piston Dome & Rod Assembly	21	
		Balance Shaft Elimination	14	
	$250		$81	$331/Engine

DOWNSIZED RESD STUDY

In reviewing the requirements for the downsized RESD, it was concluded that a 2250-lb curb-weight vehicle with an engine compartment equal to a K-body vehicle would be used. The study's results showed that a V-drive or U-drive, four-cycle Stirling engine could be designed and packaged to perform in a small-size vehicle while meeting or exceeding program objectives.

A preliminary cross section of a downsized RESD is shown in Figure 1-11. A major design concept utilized in the downsized RESD was the use of an annular heater head as opposed to the canister heater head used in the Mod I engine and full-size RESD. Figure 1-12 shows the profile of an annular heater head with one quadrant removed. Note the compact, simple construction when compared to the canister heater head (Figure 1-13), which has canister regenerators placed on an outside radius from the cylinders.

Work during the past six months has also focused on finalizing the RESD engine design, and preparing design substantiation memorandum for each component/system design. This work was done in preparation for a design review with NASA during the first half of 1983.

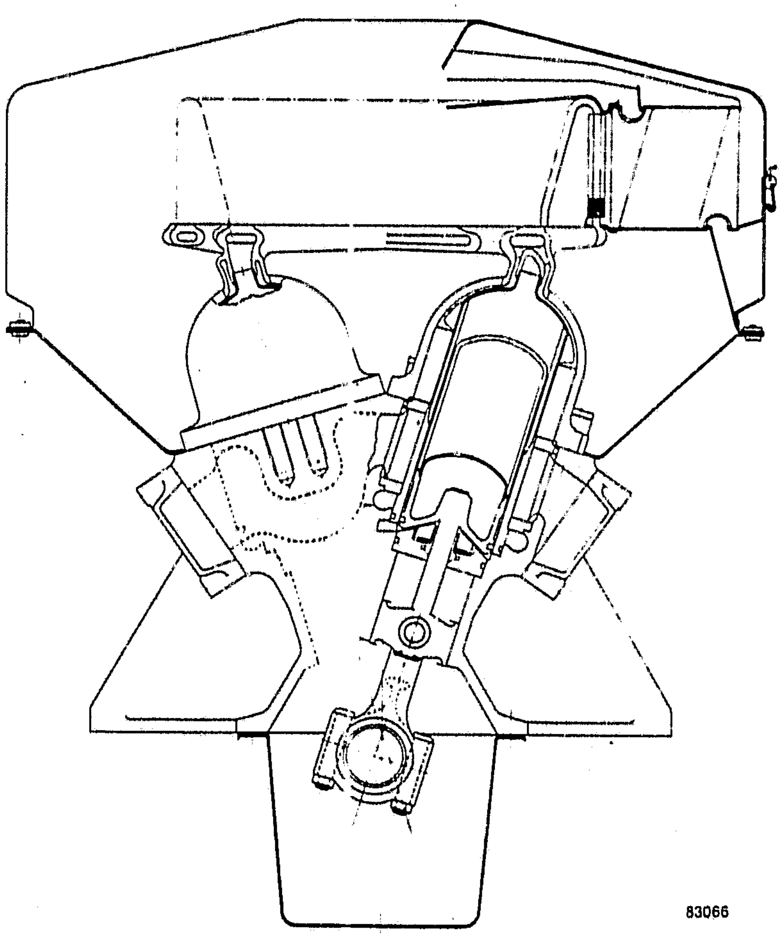

Figure 1-11 Downsized RESD

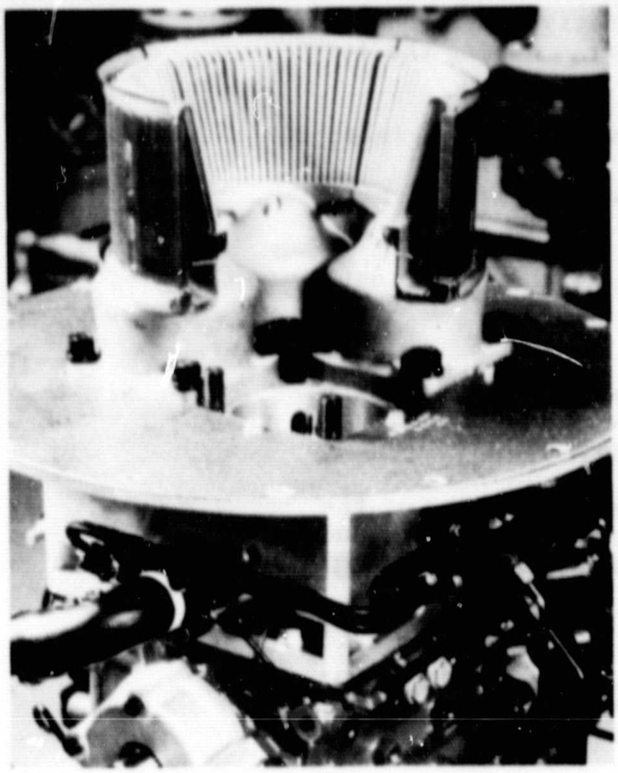

Figure 1-12 Annular Heater Head

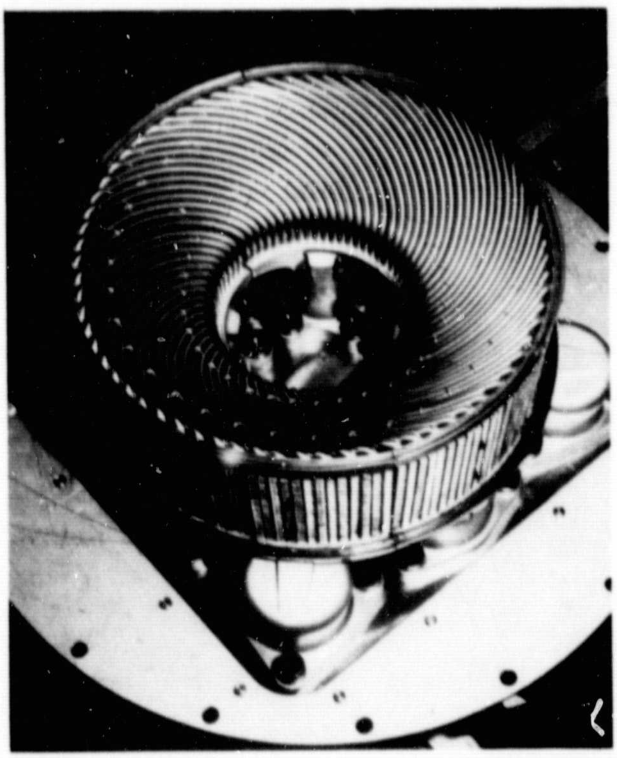

Figure 1-13 Canister Heater Head

Work Planned for Next Report Period

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The following major activities are scheduled for the ASE Program during the first half of 1983:

- assemble/test two Mod I-A engines;
- demonstrate Mod I-A engine performance and emissions goals;
- complete statistical engine testing of Mod I PL seals in three P-40 engines;
- initiate statistical engine testing of improved main seals for Stirling engines;
- complete initial test phase of the ceramic preheater segments, and initiate engine preheater design;
- complete RESD design review;
- begin component development in annular heater head design concept, and manufacture;
- complete initial piston ring development of revised H-rings;
- complete 2000-hour endurance test program on Mod I engine No. 3; and,
- complete cold-weather start program on transient test bed.

Component development activity is organized on an engine subsystem basis, with developmental emphasis on the following: 1) External Heat System (combustor, fuel nozzle, igniter, preheater); 2) Hot Engine System (heater head/regenerators); 3) Materials (heater head casting/tube materials; 4) Cold Engine System (piston ring, main seal/capseal systems, piston domes, cylinder liner); 5) Engine Drive System (crankcase, crankshaft, bearings, connecting rods); and 6) Control System (mean pressure, combustion, temperature, and microprocessor-based controls).

The focal point of development activity during this semiannual report period has been Mod I-A component development/performance characterization. This has been accomplished through substantial technology transfer and acceleration of the utilization of test facilities in the United States.

During the first half of 1983, primary emphasis will be placed on design substantiation through Mod I-A engine testing, completion of remaining Mod I-A development, and initiation of component technology development for the Mod I-B. Each component will have a specific performance, cost-reduction, or reliability goal for the Mod I-B, and will be developed to the point where they are suitable for proof-of-concept testing on a Mod I, Mod I-A, or Mod I-B engine.

External Heat System (EHS)

The primary goal of the EHS is low emissions while maintaining high efficiency for a 18:1 fuel turndown ratio in a minimum volume. The design must consider the expected use of alternate fuels, and recognize the significant cost impact of system size and design.

Development activity during the latter half of 1982 focused on the selection of a nonair-atomized Mod I-A fuel nozzle, the design of an improved CGR combustor for the Mod I-A, Performance Rig baseline testing, an analysis of Mod I engine and vehicle EGR emissions, and the development of a ceramic Mod I-A preheater.

Primary objectives during the first half of 1983 will focus on the development of a reduced size/cost EHS through rig testing, and verification of EHS performance in a Mod I-A engine. A metallic preheater will be used as an interim design until a ceramic matrix can be developed.

MOD I-A FUEL NOZZLE EVALUATION

The Mod I-A fuel nozzle must be capable of operation over a turndown range of 18, allow the engine to idle at 0.25 g/s fuel flow, not use an external source of atomizing air, and demonstrate a beneficial or neutral affect on emissions/temperature profile. Based on spray-quality evaluation and Free-Burning Rig* ignition/blowout tests, two nozzles (piloted air-blast and dual-orifice) were selected for further evaluation in the Combustion Performance Rig*.

Initially, Performance Rig evaluation was conducted using a modified (to adopt the nozzles) Bill-of-Materials (BOM) Mod I CGR combustor. Because temperature and durability problems existed with the CGR combustor, a decision was made to use an EGR-type combustor as the prime design for the Mod I-A. The combustor was modified to provide a separate igniter, and accept the nonair-atomized fuel nozzles (see Figure 2-1).

Five fuel nozzle/EGR combustor combinations were tested without EGR during this report period; these are shown in Table 2-1 and Figures 2-2 and 2-3.

*see MTI Report No. 82ASE278SA2

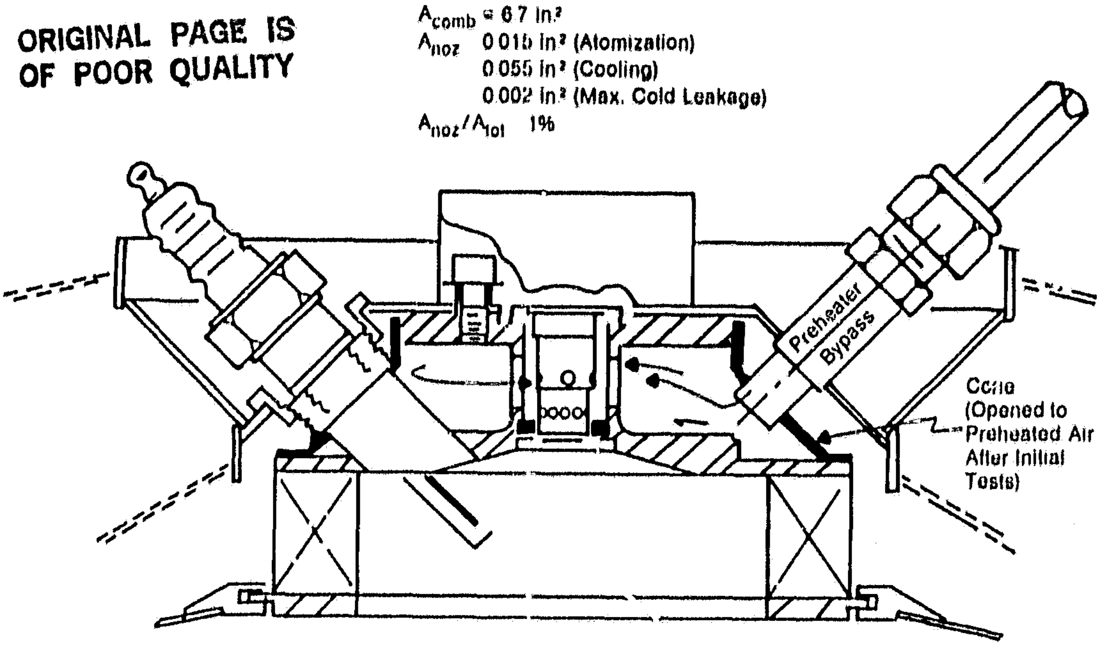

Figure 2-1 Mod I EGR - Alternate Fuel Nozzle

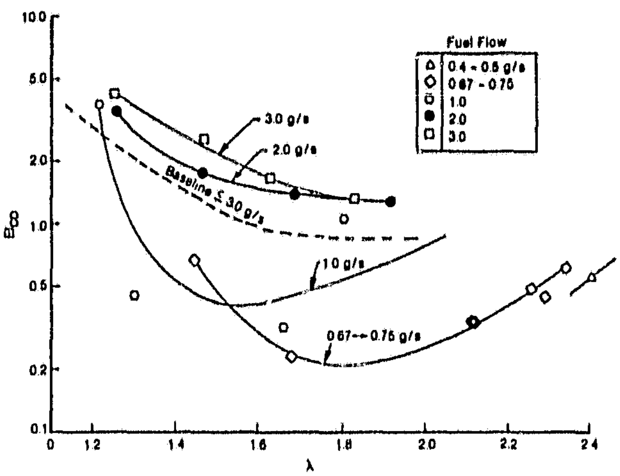

Figure 2-2 Performance Rig Comparison of Affects of Combustor Configuration on CO Emissions

TABLE 2-1

PERFORMANCE RIG EVALUATION OF MOD I-A FUEL NOZZLES

Configuration	Nozzle	Combustor
1	Air-Blast	Mod I EGR
2	Dual-Orifice	Mod I EGR
3	Air-Blast	2/3-Area Mod I EGR (Reduced Vanes)
4	Air-Blast	2/3-Area Mod I EGR (Reduced Height)
5	Air-Blast	2/3 Area with Radial Holes Mod I EGR

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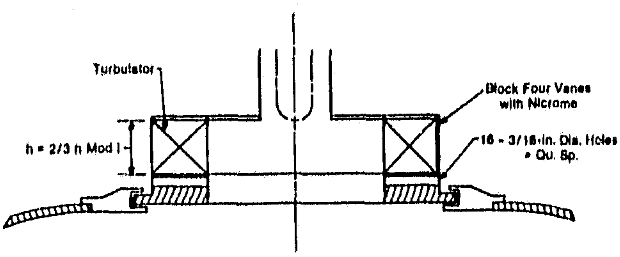

Figure 2-3 Mod I-A Fuel Nozzle Evaluation - EGR Combustor Configuration

Both fuel nozzles have a pressure-atomized pilot nozzle, but different secondaries. The air-blast secondary nozzle uses combustor pressure drop to force air through the nozzle for atomization, while the dual-orifice has a pressure-atomized secondary. In either case, combustor ΔP is used to generate cooling for the nozzle. This atomizing and/or cooling air may be preheater bypass (cold) or discharge (hot).

The first two configurations were unacceptable because not enough hot or cold air could be provided for adequate atomization/cooling, causing high CO and HC emissions, poor combustion-gas temperature distribution, and carbon formation. The remaining three configurations involved combustor modifications to increase pressure drop and, as a result, airflow through the nozzle. Based on initial testing results and previous ignition/blowout tests, a decision was made to continue this development using the air-blast nozzle.

During the third test series, the combustor flow area was reduced nominally to 2/3 of its original value by blocking every third swirl vane. Due to some warpage between the turbulator and its supporting plate, some of the air bypassed the swirler; thus, the increase in pressure drop was not as great as expected. Nevertheless, performance was markedly improved, as illustrated in Figure 2-2 where CO emissions are comparable to those achieved with the baseline Mod I EGR-combustor/air-atomized fuel nozzle.

Except for the nozzle's tendency to partially plug at very low idle fuel flows, configuration 3 was deemed acceptable for the Mod I-A. With EGR, the tendency is expected to improve greatly due to increased flow through the combustor at idle. The continued development of configurations 4 and 5 represent changes to create a more packageable/manufacturable production combustor.

Configuration 4 was intended to aerodynamically duplicate configuration 3. In this case, area reduction was achieved by reducing the height to the turbulator by 1/3; however, combustor performance was not the same, nor was it acceptable because of high heater head temperature spread. Deterioration in temperature profile was due to an increase in swirl intensity which resulted from elimination of the leakage path underneath the turbulator. Configuration 5 corrected this by adding sixteen 3/16-diameter radial holes under the turbulator (see Figure 2-3), resulting in performance comparable to configuration 3.

The EGR combustor, represented by configuration 5 and the air-blast nozzle, was selected for incorporation into the Mod I-A. A test to verify performance and evaluate the affect of EGR will be conducted on Mod I engine No. 10 during January, 1983. Additional nozzle development using other air-blast/dual-orifice nozzles, as well as an ultrasonic atomizer, will continue throughout 1983 if further nozzle improvement is needed.

MOD I-A CGR COMBUSTOR

Although the initial combustor for the Mod I-A will be an EGR design, efforts have continued to develop a CGR combustor that can be retrofitted at a later date. The purpose of both gas-recirculation schemes is to ensure CVS cycle emissions compliance. The combustor must also minimize gas temperature spread to ensure high engine efficiency, operate with low excess air and pressure drop, be stable at off-design conditions, and be durable. The CGR concept is desirable because of its inherent simplicity, e.g., lack of external valving, piping, and controls.

Development of both tubular and radial CGR combustors continues. The initial tubular combustor was modified to improve durability and simplify the design, as shown in Figure 2-4. Durability improvement results from suspending the tubes at the fuel nozzle by a weld joint, and not constraining them at the ejector; thus, tubes can move relative to the rest of the combustor and EHS during thermal transients. The tube design was modified so that the cross-sectional geometry would change from circular (ejector side) to rectangular (nozzle side). The rectangular tubes are joined together to form a turbulator and, therefore, eliminate the support ring that would otherwise be needed.

Mod I engine testing of the tubular CGR combustor with 13-mm diameter ejectors revealed a poor heater head temperature profile, high NO_x emissions (Figure 2-5), and acceptable CO (Figure 2-6) and smoke emissions. Based on the NO_x and temperature profile, a decision was made to design the Mod I-A tubular CGR combustor with circular tubes and smaller diameter ejectors in order to capitalize on the good performance achieved previously on the Mod I with this type of design.

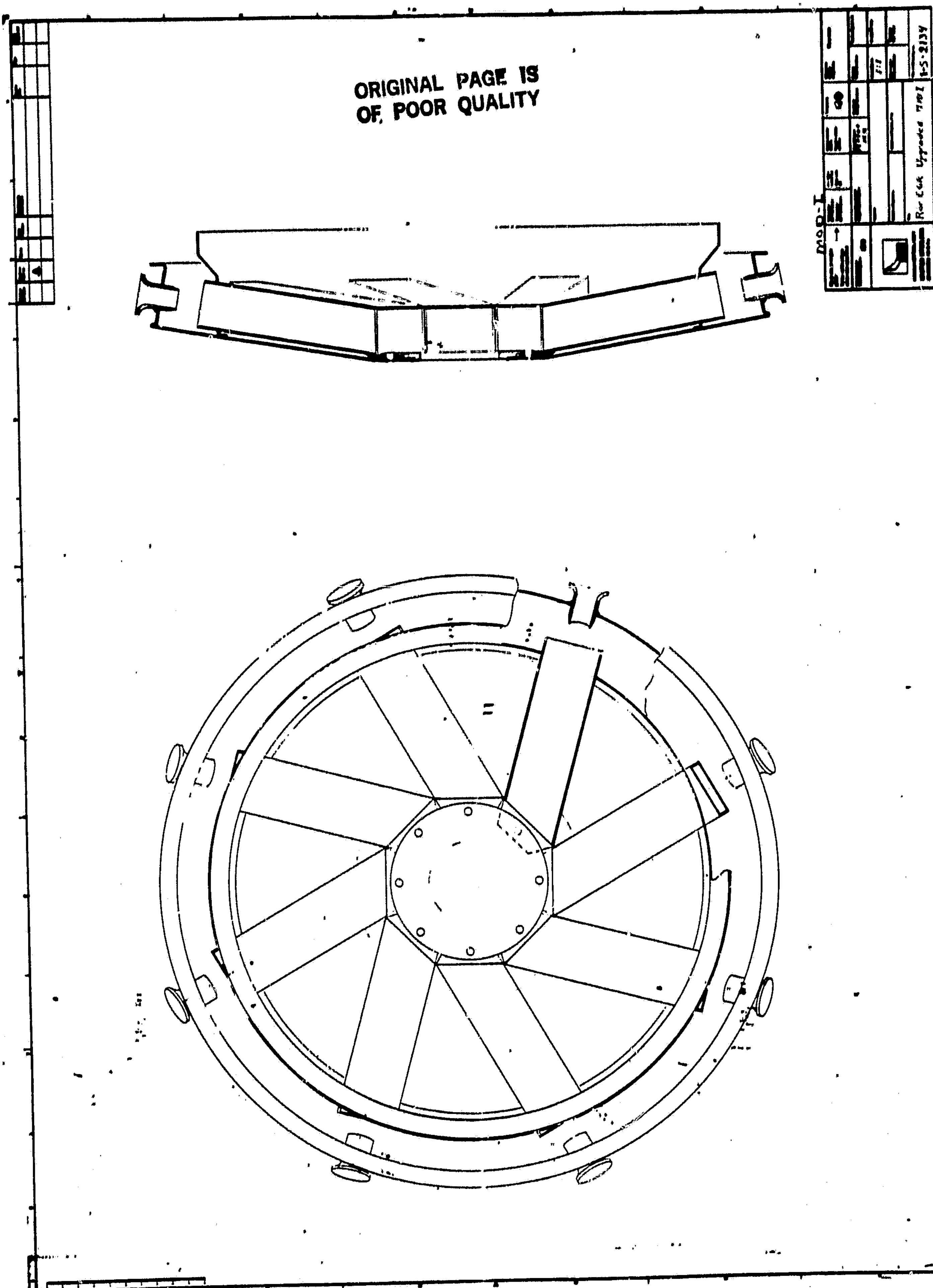

Figure 2-4. USAF Improved Tubular CGB Combustor

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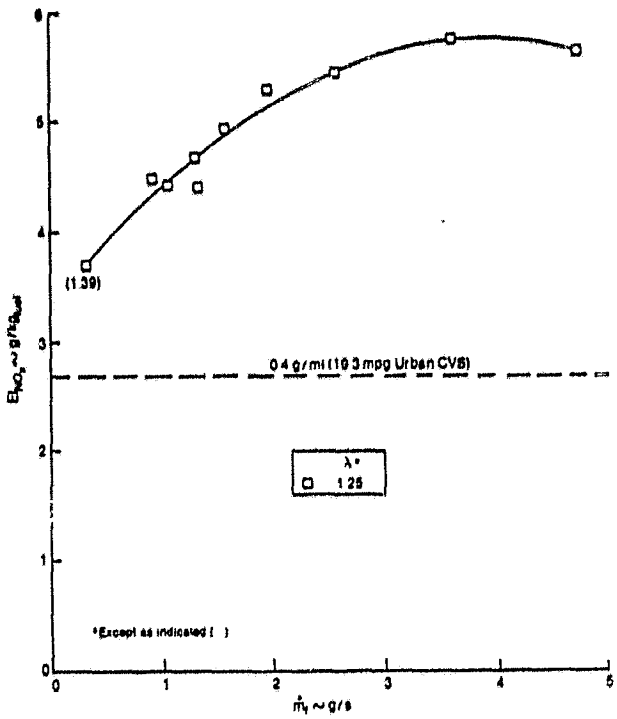

Figure 2-5 Mod I Engine No. 2 NO_X Emissions with CGR

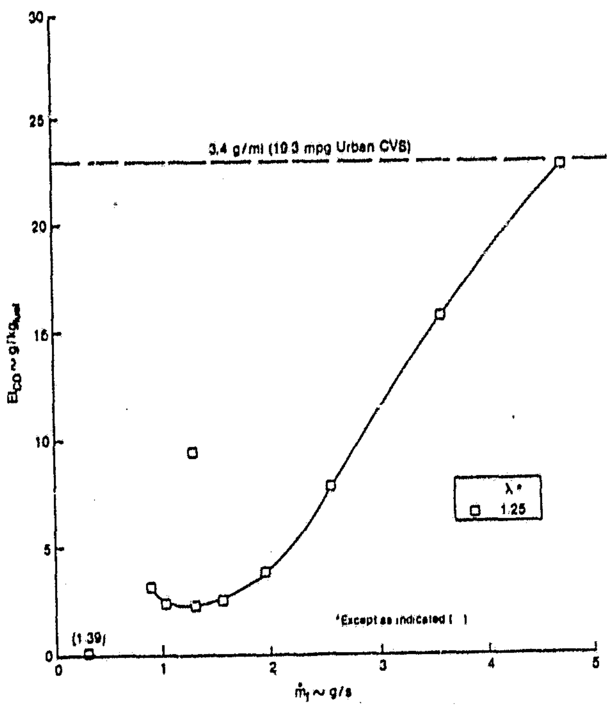

Figure 2-6 Mod I Engine No. 2 CO Emissions with CGR

Development of the radial CGR combustor was delayed because of the emphasis placed on Mod I EGR emissions, and the decision to develop Mod I-A fuel nozzles with an EGR combustor, as opposed to a straight-guide-vane CGR combustor. Hardware for these radial CGR combustors with varying geometry has been received and assembly has begun; development in the Performance Rig will begin early in 1983.

PERFORMANCE RIG MOD I BASELINE TESTS

The Combustion Performance Rig was brought to full operational capability during the latter half of 1982, and two series of Mod I baseline tests with CGR and EGR combustors were completed.

The Data Acquisition System (DAS) was fully automated during this report period using the HP-1000 computer. A dramatic increase in test points run per hour resulted.

The first series of baseline tests used the Mod I BOM, straight-guide-vane, CGR combustor and air-atomized fuel nozzle. With the CGR bypass valve welded closed, tests were conducted at fuel flows of 0.5, 1.0, 2.0, and 3.0 g/s, and Lambda's (λ) beween 1.18 and 2.00 in order to demonstrate the capabilities of the rig and compatability between MTI/USAB rig data; both objectives were achieved.

In the latter case, a comparison of rig data for preheater effectiveness (see Figure 2-7), NOx emissions (see Figure 2-8) and CO emissions (see Figures 2-9 and 2-10) indicates similar trends and qualitative agreement. Differences are due to variations in the methods of controlling tube temperature, and the inherent sensitivity of the straight-guide-vane CGR combustor to manufacturing tolerances.

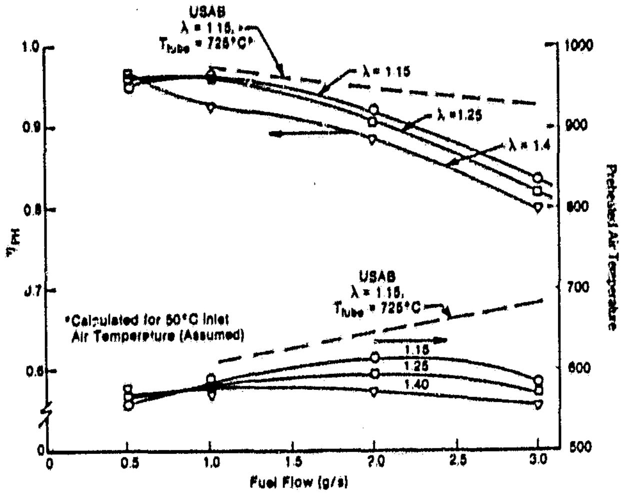

Figure 2-7 Mod I Baseline Test (8/5-6/82) - Preheater Effectiveness Air Temperature Versus Fuel Flow

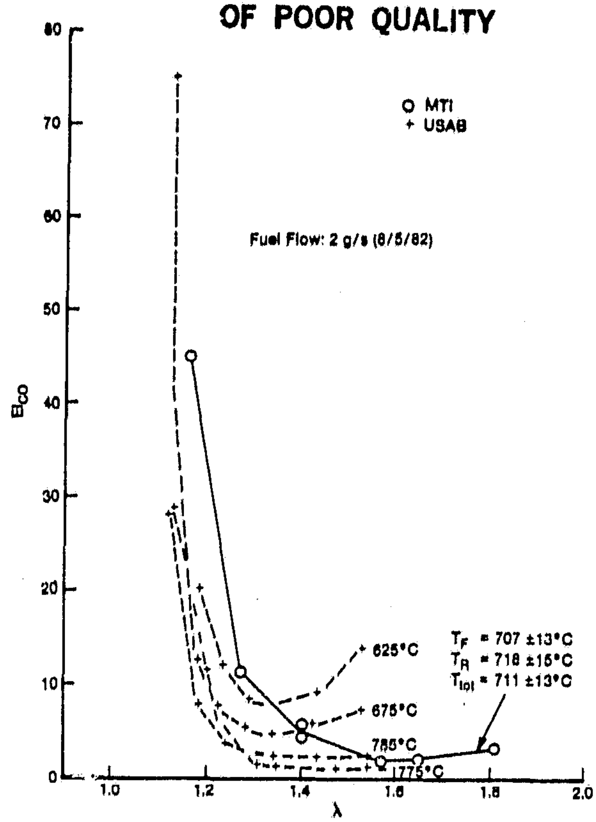

Figure 2-9 Mod I Baseline CO Versus λ

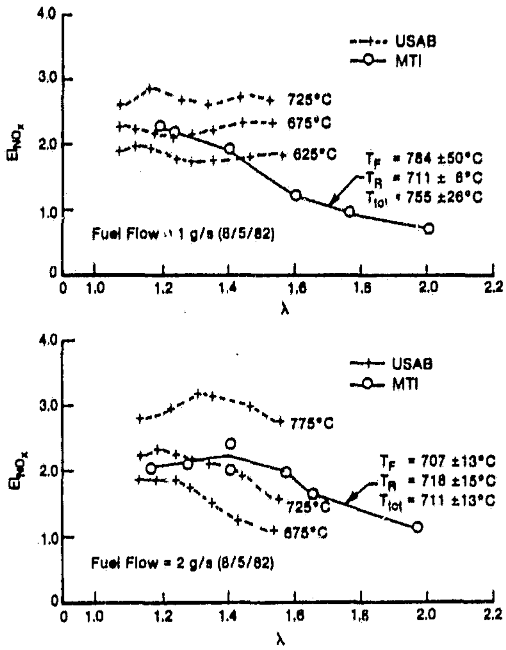

Figure 2-8 Mod I Baseline NO_X Versus λ

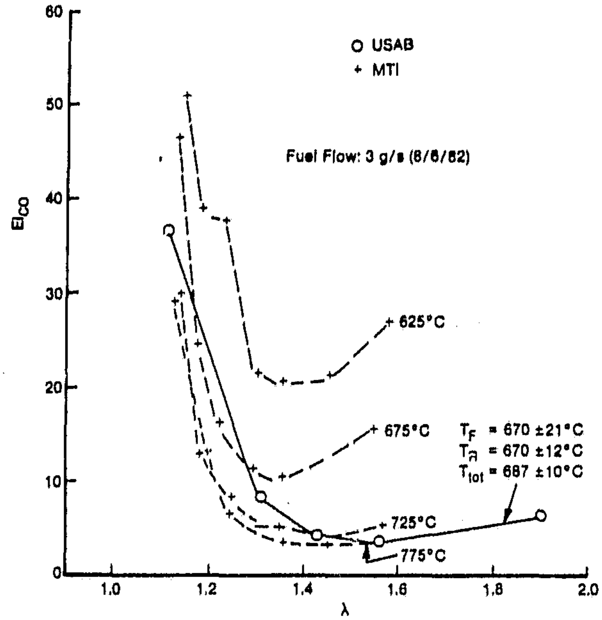

Figure 2-10 Mod I Baseline CO Versus λ

The second series of baseline tests were conducted to establish criteria for Mod I-A fuel nozzle selection. In order to compare nozzle performance with an EGR combustor, a baseline was run using the Mod I EGR combustor and air-atomized fuel nozzle. The results of the test were used to select the Mod I-A fuel nozzle, and to determine the degree of EGR combustor modification needed to utilize that nozzle as previously discussed.

MOD I ENGINE/VEHICLE EGR EMISSIONS

Emissions testing was conducted with EGR using both the engine cell and vehicle. A detailed comparison between steady-state engine and transient CVS cycle vehicle data yielded excellent agreement for NO_x, but not for CO or HC emissions; however, good engine-to-engine repeatability was demonstrated. During this report period, gas analyzer data acquisition and calibration procedures were fully automated with the HP-1000.

Data from a limited emissions test conducted with EGR using Mod I engine No. 1 was compared to that obtained with other engines. NO_x emissions of two Mod I engines (versus fuel flow) with variable EGR schedules are shown in Figure 2-11. The variation of EGR and λ with fuel flow is given in Figure 2-12. It was concluded that variations in λ had no affect on NO_x and that good engine-to-engine repeatability with EGR was demonstrated (see Figure 2-13 where NO_x emissions from three Mod I engines - one constant and two variable EGR schedules - are illustrated).

CO emissions (shown in Figure 2-14) seem to indicate an effect of fuel flow, EGR, and λ, which is the dominant variable (see Figure 2-15). Other conclusions drawn are that good engine-to-engine repeatability exists, and CVS cycle CO compliance requires λ > 1.2; HC emissions were so low in all cases that comparison of engine data was not relevant.

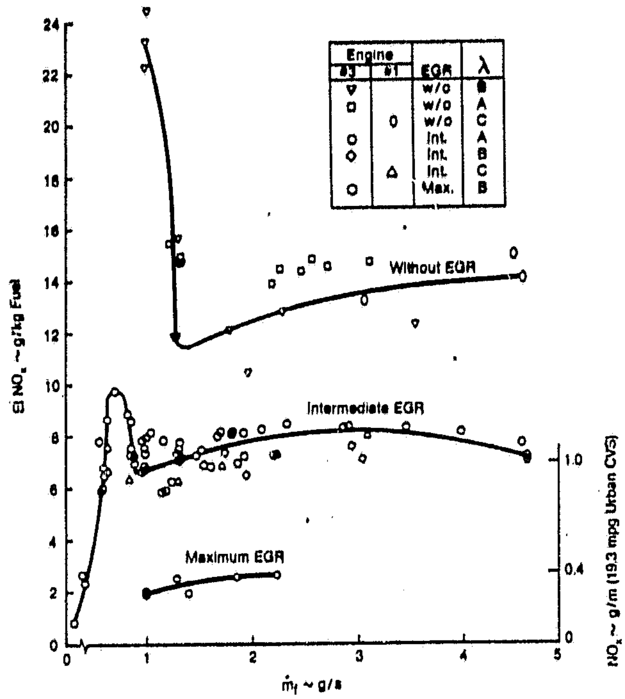

Figure 2-11 Mod I Engine NO_x Emissions With/Without EGR

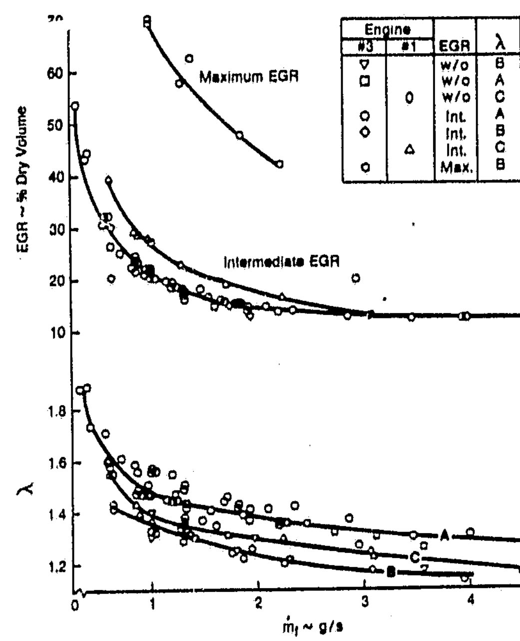

Figure 2-12 Mod I Engine Variation in λ and EGR With Fuel Flow

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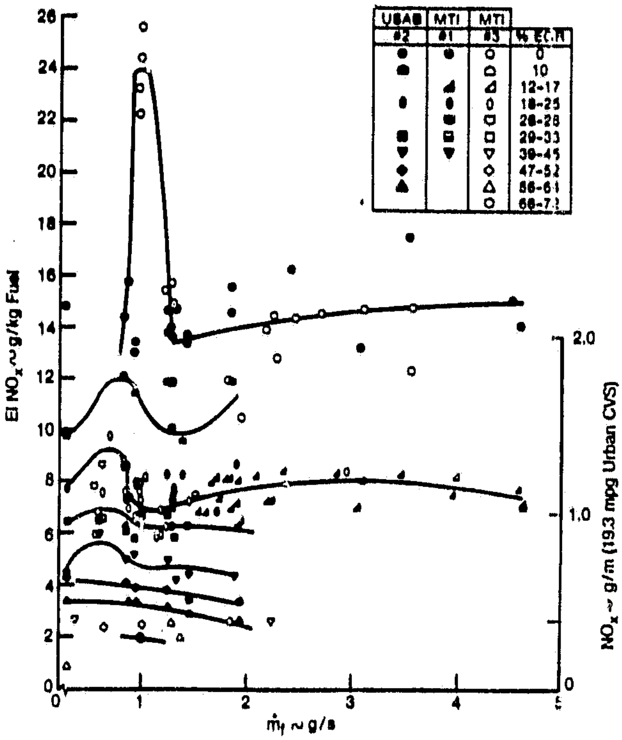

Figure 2-13 Mod I Engine NO_X Emissions With/Without EGR

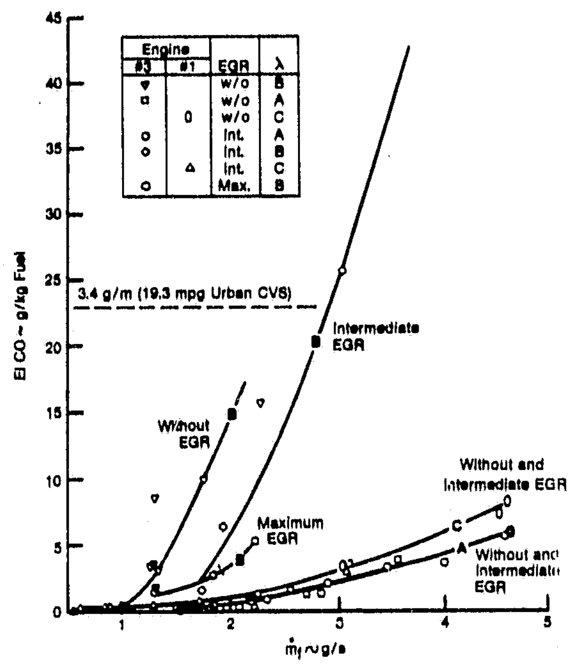

Figure 2-14 Mod I Engine CO Emissions With/Without EGR

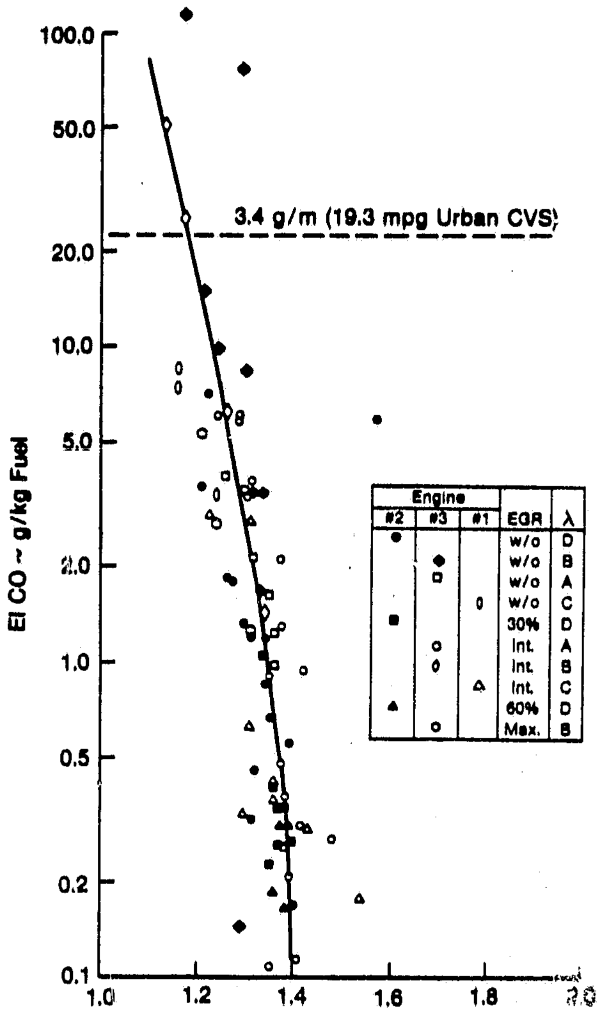

Figure 2-15 Effect of λ on Mod I Engine CO Emissions

Following the testing of Mod I engine No. 1, the engine was installed in the Lerma vehicle, and CVS cycle tests were conducted at Mercedes Benz of Ann Arbor, Michigan. The EGR combustion system was the same as that tested in the engine cell with the "intermediate" EGR schedule shown in Figure 2-12. Based on the engine results projected over the CVS cycle, and using an urban mileage of 19.3 mpg, emissions were predicted to be:

- NO_x = 0.88 g/mi;
- CO < 0.50 g/mi (λ > 1.2); and,
- HC < 0.10 g/mi.

The first series of CVS vehicle tests showed that NO_x emissions were about the same as predicted (1.0 g/mi); particulate emissions were well under the required 0.2 g/mi. CO and HC emissions were an order of magnitude higher than expected, thus exceeding the Program goals.

Burning diesel fuel instead of indolene (urban cycle without cold start) indicated a twofold increase in CO, and a threefold increase in HC. A subsequent analysis revealed: λ too rich (< 1.2), excessive servo-oil temperature and, air throttle setting incorrect for cold start. Due to the dramatic affect of λ on CO emissions, the test was repeated with a more optimum λ schedule (results are shown in Table 2-2). NO_x emissions were almost as predicted, and the use of steady-state engine data to predict CVS cycle vehicle NO_x emissions was validated. CO and HC emissions satisfied the Program goals of 3.4 and 0.41 g/mi, respectively; however, they were higher than predicted due to the affect of start-up and transients. As a result, it is concluded that steady-state CO and HC emissions cannot be used to predict vehicle results.

TABLE 2-2

MOD I TRANSIENT TEST RESULTS

FTP-79 Cold Urban Cycle				
Test Date	HC (g/mi)	CO (g/mi)	NO_x (g/mi)	Fuel Economy (mpg)
9/21/82	0.225	3.43	0.96	19.90
9/24/82	0.286	3.27	0.90	18.80
9/25/82	0.247	3.21	0.84	19.20
Average	0.253	3.30	0.90	19.30
Projected	<0.100	<0.50	0.88	19.09

Highway Cycle					
Test Date	HC (g/mi)	CO (g/mi)	NO_x (g/mi)	Fuel Economy (mpg)	Combined Fuel Economy (mpg)
9/21/82	0.0040	0.280	0.750	31.70	23.90
9/24/82	0.0050	0.250	0.620	32.10	23.15
9/25/82	0.0040	0.402	0.611	32.40	23.50
Average	0.0043	0.311	0.660	32.10	23.50

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MOD I-A CERAMIC PREHEATER

The metallic preheater represents one of the most expensive components of the engine due to the inherent expense of fabricating individual metallic platelets into a matrix. In order to reduce this cost, ceramic material, which is relatively inexpensive to fabricate, has been under evaluation for use in the preheater. In addition to cost considerations, ceramic preheater performance and thermal mass must be consistent with high EHS efficiency, minimal cold-start penalty, and packageability. These requirements imply extremely high surface/volume ratio (300 ft^2/ft^3), thin walls, and low heat capacity (mass times specific heat).

The ceramic-preheater concept selected for development is shown in Figure 2-16. Two 4-3/4" x 4-1/2" x 3-3/4" cordierite blocks were purchased for performance/durability evaluation in the Preheater Rig*. These counterflow blocks, originally developed for use as a gas-turbine recuperator, promise to be less expensive than the current metal platelet preheater, give comparable performance, and reduce thermal mass by as much as 30%.

The Preheater Rig used to evaluate the ceramic blocks supplies the test section with both a hot and cold air stream; the hot stream may be either electrically heated air (up to 500°C and low flows), or the product of natural-gas combustion (> 500°C). The evaluation will consist of two phases. During the first phase, which will include pressure drop, temperature profile, and effectiveness, steady-state performance will be determined at 300, 500, 700, and 900°C with varying flow rates. If the test pieces survive the high-temperature, steady-state tests, they will be subjected to thermal transients in order to determine durability in an engine environment. This performance evaluation has been initiated, and tests conducted at 300 and 500°C with preheated air.

Cordierite is not expected to have satisfactory durability for an engine environment. If successful performance and cost benefits are demonstrated, additional materials and geometries will be evaluated in the Preheater Rig to develop a ceramic-block preheater for the Mod I-A or RESD engines. The performance, durability, and cost evaluation will be completed during the first half of 1983.

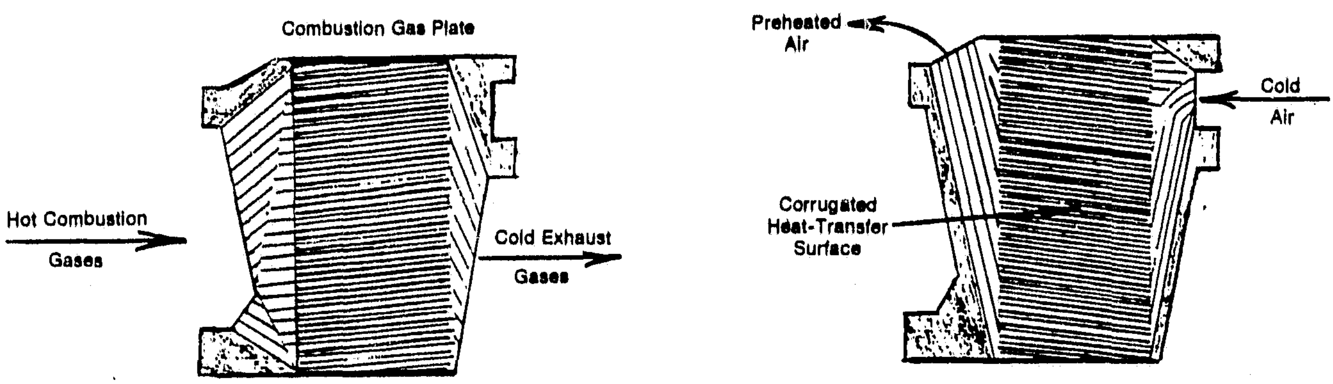

Figure 2-16 Ceramic Preheater Test Section

*completed and operational in late 1982 (see MTI Report No. 82ASE278SA2)

Analytical work during this semiannual report period included the development of a preheater computer code that will utilize data from the Preheater Rig to optimize engine ceramic-preheater heat transfer, fluid friction, and cold-start penalty. An example of the results of this code is shown in Figure 2-17 where the affect of the number of ceramic plates on ceramic RESD preheater effectiveness and pressure drop is indicated and compared with projected Mod I Performance.

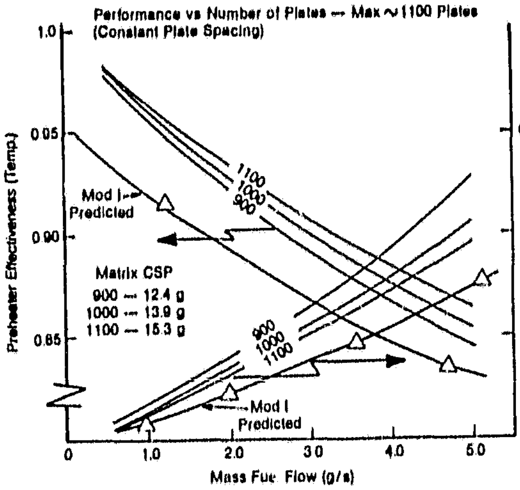

Figure 2-17 Ceramic Preheater Performance (Preliminary)

Hot Engine System Development

The primary goal of this task is to provide a low-cost regenerator design with mileage performance comparable to the current design. To date, the requirement for high regenerator efficiency (typically ~98%) has been reflected in the use of high-cost matrix material such as sintered woven, high-count stainless mesh.

Activity during this report period was aimed at rig development of ceramic-regenerator matrix material, and evaluation of Metex knitted wire in a Mod I engine.

CERAMIC-REGENERATOR DEVELOPMENT

Performance evaluation of the SiC Duocel matrix has been completed in the Heat-Transfer Rig with encouraging results. Four porosities were tested (see Figures 2-18 and 2-19). Both heat-transfer and friction-factor characteristics improve as porosity increases, and are comparable or better than the currently used Mod I sintered wire screen. Given the possibility of an order-of-magnitude reduction in material cost, SiC is an attractive alternative to sintered screen, and in the prime design for the RESD.

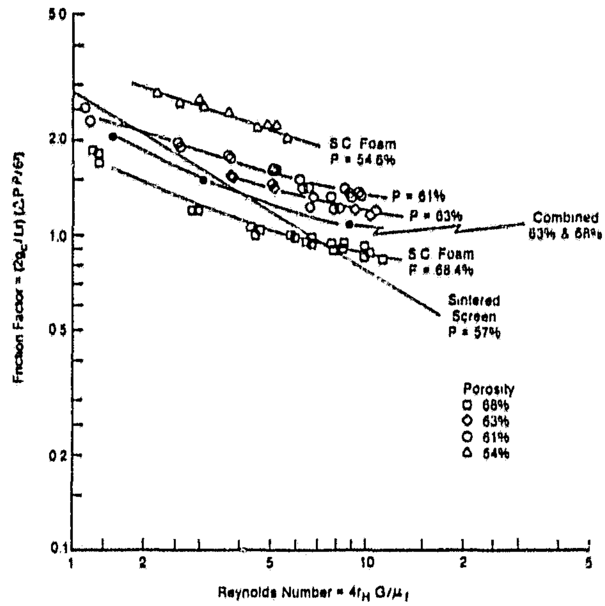

Figure 2-18 Nusselt Number Vs. Reynolds Number of Duocel Regenerator Matrix

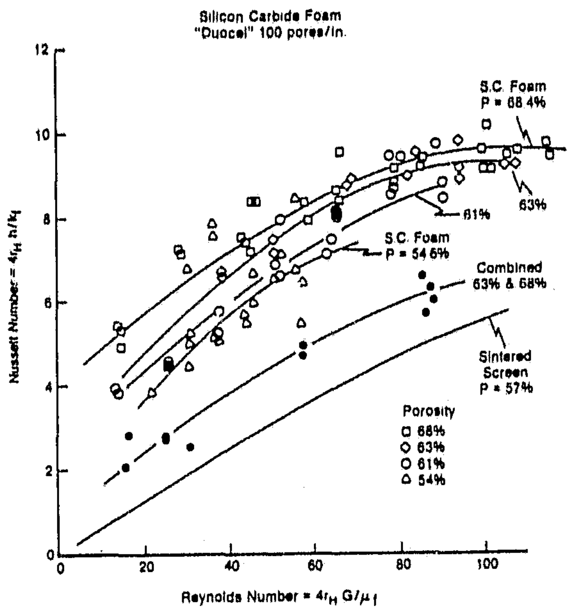

Figure 2-19 Friction Factor Vs. Reynolds Number of Duocel Regenerator Matrix

Performance characteristics of SiC are not anticipated to be as good in the engine due to its high axial heat conduction. In order to get a more representative comparison of rig data, which assumes no axial conduction, an additional test was run using combined 63%-/68%-porosity SiC separated by an insulating ceramic. With greatly reduced axial conduction, a direct comparison can be made to the sintered wire screen baseline. As shown in Figures 2-18 and 2-19, heat-transfer performance of the combined insulated SiC was not as good as the single uninsulated porosities, while, as expected, friction factor was unaffected. Nevertheless, performance is still superior to sintered screen.

The major question remaining, considering SiC's lower cost and superior performance, is durability. To evaluate the life characteristics in a cyclic environment, a Regenerator Durability Rig was constructed using a diesel-engine block to cycle nitrogen through a test matrix. To date, the SiC material has survived nine hours of rig testing, representing over one million pressure cycles without damage. The SiC regenerator will be evaluated in a P-40 engine during the first half of 1983.

METEX MOD I ENGINE REGENERATOR

Metex knitted-wire regenerators have been designed, fabricated, and flow-tested for the Mod I engine in 55% and 70% porosities. Based on Heat-Transfer Rig data, and analysis by the First-Order Code, a 70% porosity is predicted to be optimal.

Testing of the regenerators in a Mod I engine has been delayed because of fabrication difficulties and engine availability. By comparing the engine performance with sintered screen, 70%-/55%-porosity Metex regenerators, rig data and engine code predictions can be validated. The 70%-porosity Metex regenerators have been installed in Mod I engine No. 10, with testing scheduled for January, 1983.

Materials and Process Development

The main objective of this task is the utilization of low-cost, nonstrategic heater head materials that can survive the automotive duty cycle. The high-temperature/pressure environment, as well as the presence of high- and low-cycle, cyclic stresses, have contributed to the difficulty of meeting this objective.

Development efforts during this report period focused on Phase II testing of two specimens of alternate casting materials and five alternate tube materials; reducing the strategic-element content of the Mod I-A engine; selection of material for the Mod I-A heater head, engine-testing two heater heads from these and other materials and developing data on welded XF-818.

During the first half of 1983, Phase II testing will be completed, alternate-material heater heads will be destructively tested on an engine in a simulated duty cycle, fatigue tests of the casting materials will be performed in a hydrogen environment, and alternate techniques of reducing hydrogen permeability will be pursued.

DESIGN PROPERTIES TESTING

The data scatter of fully reversed, high-temperature fatigue testing of alloys XF-818 and CRM-6D was demonstrated during the last semiannual report period. Initial Phase I testing was done at 800°C. In order to observe the data scatter and develop 700°C fatigue data, six tests were run at low-cycle and six at high-cycle for each material (Figures 2-20a and b show the results).

Results of testing currently underway to determine the mean stress and hold time effects of XF-818, and to determine the fatigue strength in hydrogen of XF-818 and CRM-6D, will be reported during the first half of 1983.

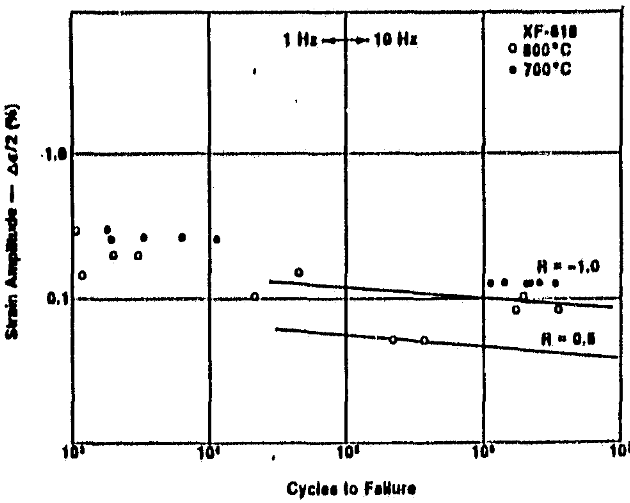

Figure 2-20a Phase II, Step I, 700°C
Statistical Significance (XF-818)

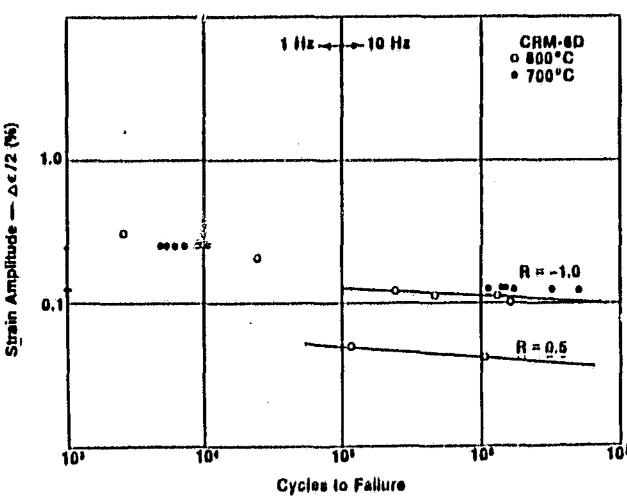

Figure 2-20b Phase II, Step I, 700°C
Statistical Significance (CRM-6D)

Tensile strengths of welded XF-818 have been determined at 800°C. The test bars contained a weld in the center of the gage length which simulated the actual welding process in both electron-beam (EB) and TIG welding. Two different filler metals were used with TIG welding, while none was used with EB welding. The results, shown in Table 2-3, are compared to initial Phase I tensile data, to tensile properties reported by Climax Molybdenum, and to the initial test bars cast by Howmet in preparation for casting of the Mod I-A heater head castings.

Tensile properties of the welded specimen appear excellent, perhaps indicating a small trend for the braze or HIP cycle to slightly decrease yield strength at both room temperature and 800°C while possibly increasing ductility.

During the first half of 1983, current fatigue testing of welded XF-818 will be reported, along with hydraulic fatigue testing of Mod I-A heater head castings. These tests will verify the basic integrity of the design/alloy combination at room temperature, and will be run at 15 MPa ±5 MPa for 10^7 cycles.

Long-term tests (to 5000 hours) are being conducted on the five alternate heater tube alloys (Table 2-4 lists these alloys and their compositions). The latest 850°C test data from the alternate alloys is plotted are Figures 2-21 through 2-25. The stress to rupture at 850°C at 3500 hours has been estimated from the graphic data and summarized below:

Alloy	Estimated Stress N/mm^2
CG-27	51
Inconel 625	44
12RN72	35
Multimet	32
Sanicro 31H	27
Sanicro 32	24

The design stress, based on an equivalent operational pressure of 7.5 MPa and a safety factor, is 28.125 N/mm^2. Considering this, Sanicro 31H and 32 will not meet the requirements; all the other alloys appear to meet the requirement for creep rupture at this time.

TABLE 2-3

TENSILE TEST RESULTS OF WELDED XF-818

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	Room Temperature				800°C			
	TS	0.2% YS	% EL	Remarks	TS	0.2% YS	% EL	Remarks
TIG-Welded Hastelloy With Filler	95.84	68.43	1.90	2, 3, 6	70.235	45.306	4.00	2, 3, 6
TIG-Welded XF-818 Filler	96.91	67.01	1.87	2, 3, 7	69.25	43.79	4.10	2, 3, 7
EB-Welded No Filler	98.23	66.24	2.76	2, 3, 8	65.69 68.38	39.51 36.32	5.60 5.25	2, 3, 6 2, 7, 10
Climax Molybdenum	97.40	59.60	1.60	1, 4	61.60	—	9.00	1, 11
Phase I (B.K.)	77.50	45.25	1.40	1, 12, 6	62.00	9.00	5.30	1, 7, 12
Howmet as Cast	97.20	64.10	2.30	4, 6	—	—	—	
Howmet HIP'ed	97.40	54.80	4.00	5, 9	—	—	—	

Remarks:
1. Investment-cast test bars
2. Test bars machined from welded investment-cast slabs
3. Aged 50 hours at 800°C
4. As cast
5. HIP'ed - 1121°C; 15 ksi 4 hours
6. Average 2 specimens
7. 1 Specimen
8. Average 3 specimens
9. Average 4 specimens
10. 1121°C - 1 hour; 800°C - 50 hours
11. 815°C test temperature
12. 1125°C - 1 hour; 800°C - 50 hours

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TABLE 2-4

ALTERNATIVE HEATER TUBE MATERIALS, NOMINAL CHEMISTRY

Alloy	Co	Cr	Ni	Mo	W	C	Al	Ti	B	Cb	Mn	Fe	Si	N
Multimet[1] (N155)	19.75	21.25	20	3.00	2.5	0.12	—	—	—	1.0	1.50	29.70	1.0	0.15
Alternatives:														
CG-27[2]	None	13.00	38	5.75	—	0.05	1.6	2.5	0.010	0.7	—	38.00	—	—
Inconel 625[3]	None	21.50	61	9.00	—	0.05	0.2	0.2	—	—	0.25	2.50	0.2	—
Sanicro 32[4]	None	21.00	31	—	3.0	0.09	0.4	0.4	—	—	0.60	42.80	0.6	—
Sanicro 31H[4]	None	21.00	31	—	—	0.07	0.3	0.3	—	—	0.60	46.13	0.6	—
12RN72[4]	None	19.00	25	1.40	—	0.10	—	0.5	0.006	—	1.80	51.80	0.4	—

[1]Base Material
[2]Crucible Steel Corporation
[3]International Nickel
[4]Sandvik Alloys

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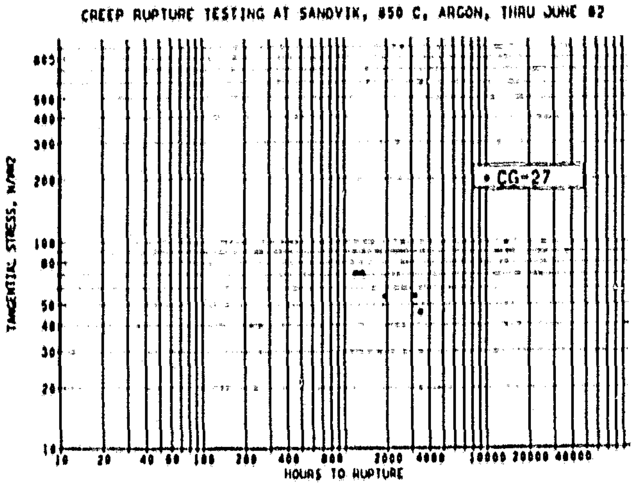

Figure 2-21 Creep Rupture Testing at Sandvik (CG-27)

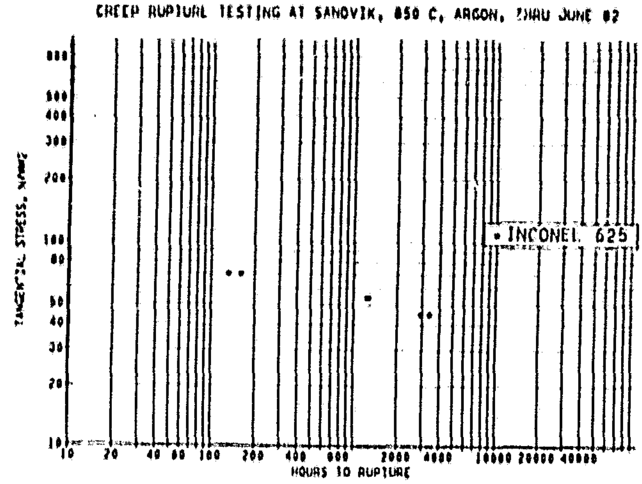

Figure 2-22 Creep Rupture Testing at Sandvik (Inconel 625)

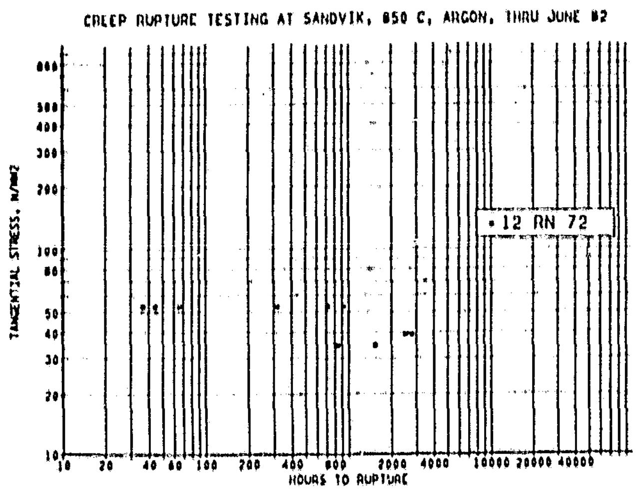

Figure 2-23 Creep Rupture Testing at Sandvik (12RN72)

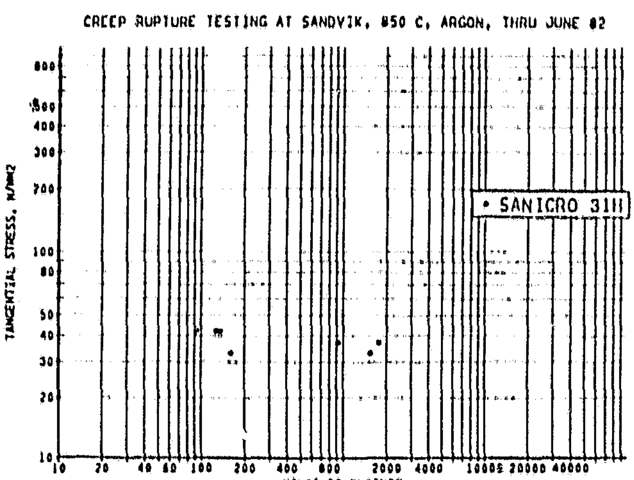

Figure 2-24 Creep Rupture Testing at Sandvik (Sanicro 31H)

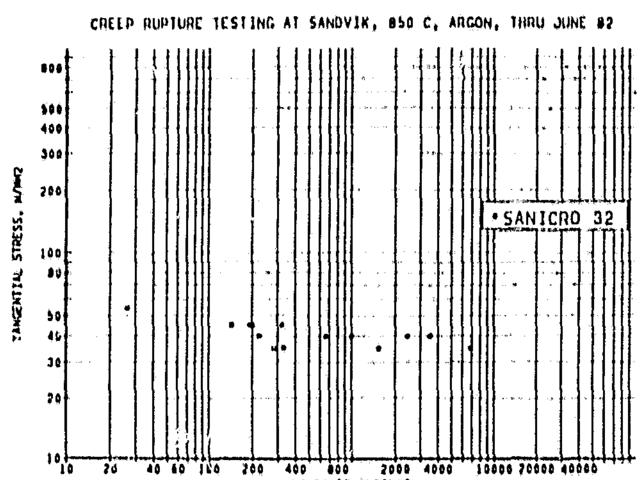

Figure 2-25 Creep Rupture Testing at Sandvik (Sanicro 32)

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HIGH-TEMPERATURE ENGINE TESTING

The alternate heater head casting and heater tube materials are being subjected to high-temperature engine testing on a P-40 engine. The objective of this test is to rank the alternate materials in an engine environment. Eight quadrants have been manufactured with the combinations of casting and tube materials shown in Table 2-5. To date, 1,453.52 test hours have been accumulated against the test goal of 2000 test hours.

The running history of each quadrant is shown in Figure 2-26. The quadrants are currently undergoing nondestructive examination. The results will help formulate the last 500-hour test cycle pressure levels. The chart in Figure 2-26 indicates that three tubes of 12RN72 in Quadrant 3 failed in about an 80-hour period. As a result, all of the 12RN72 tubes were replaced with Inconel 625.

TABLE 2-5

TUBE/CASTING MATERIALS

Quadrant No.	Material	
	Casting	Tube
1	HS-31	Inconel 625
2	CRM-6D	Sanicro 32
3	XF-818	12RN72
4	SAF-11	CG-27
5	HS-31	Sanicro 32
6	CRM-6D	Inconel 625
7	XF-818	12RN72
8	SAF-11	Sanicro 31H

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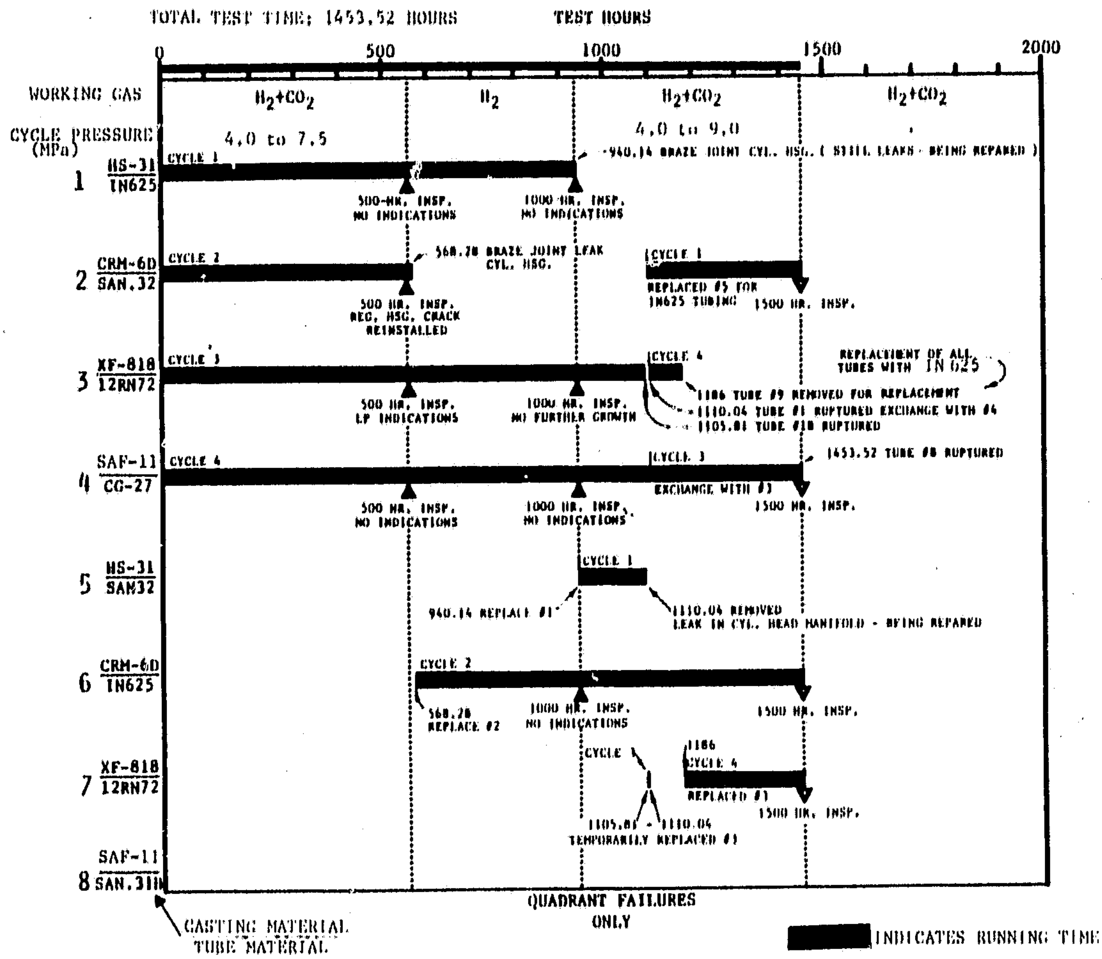

Figure 2-26 HTP-40 Log

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The alloys' nominal compositions are listed in Table 2-6. The engine is running with a mean tube temperature of 820°C, and mean pressure cycled every 90 seconds to accumulate creep and fatigue damage in the castings and tubes.

During the next report period, the last 500 test hours will be run, and the test cycle will be chosen so that the weaker alloys will fail, thus providing a ranking.

RESD MATERIALS AND PROCESS DEVELOPMENT

Work is underway to establish a materials specification for the main piston rod seal material. The specification will allow material to be purchased with a certain degree of uniformity from batch to batch, and with specific minimum-quality requirements. Testing will focus on the mechanical properties and degree of crystallinity of the elastomeric materials.

Two alternate materials (Armco and Alleghany Ludlum) for use in the air preheater are currently being investigated. Their compositions are shown in Table 2-7. Note the reduced chromium content when compared to that of 25% in the current 310 Stainless Steel.

TABLE 2-7

ALTERNATIVE PREHEATER MATERIALS

	C	Mn	Si	Cr	Ni	Al	Ti	Cb
18SR	0.10	1.0	0.9	17.5	1.0	2.2	0.45	—
408Cb	0.02	—	0.5	12.0	—	1.2	0.30	0.6

A set of samples is being exposed to the combustor environment to initially gage their ability to withstand the oxidation and corrosion atomosphere. Other tests for weldability/formability are in progress to determine the process sequence (and therefore the cost) of making an air preheater from either of these alloys.

TABLE 2-6

NOMINAL CHEMISTRY OF HEATER HEAD CASTING ALLOYS AND HEATER TUBE MATERIALS

Heater Head Casting Alloys

Alloy	C	Si	Mn	Ni	Cr	Mo	W	Co	Cb	B	N	Fe
HS-31	0.05	0.05	0.05	10.5	25.5	—	7.5	54.0	—	—	—	1.00
CRM-6D	1.05	0.55	4.75	5.0	21.0	1.0	1.0	—	1.0	0.005	—	64.5
SAF-11	0.06	0.07	0.07	16.0	24.0	—	13.0	—	—	0.045	—	44.5
XF-818	0.02	0.03	0.15	18.0	18.0	7.5	—	—	0.4	0.007	0.12	54.0

Heater Tube Materials

Alloy	Co	Cr	Ni	Mo	W	C	A1	Ti	B	Cb	Mn	Fe	Si	N
CG-27	—	13.00	38	5.75	—	0.05	1.6	2.5	0.010	0.7	—	38.00	—	—
Inconel 625	—	21.50	61	9.00	—	0.05	0.2	0.2	—	—	0.25	2.50	0.2	—
Sanicro 32	—	21.00	31	—	3.0	0.09	0.4	0.4	—	—	0.60	42.80	0.6	—
Sanicro 31H	—	21.00	31	—	—	0.07	0.3	0.3	—	—	0.60	46.13	0.6	—
12RN72	—	19.00	25	1.40	—	0.10	—	0.5	0.006	—	1.80	51.80	0.4	—

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Cold Engine System (CES) Development

The focus of CES activity is the development of reliable, low-friction reciprocating seals that have long life in an automotive duty cycle. Both main seals (piston rod) and piston rings have been targeted for improvement.

During the latter half of 1982, efforts were directed toward the development of low-friction piston rings in a Mod I Motoring Rig, determination of PL advanced-design main seal performance in a one-cycle Exploratory Test Rig, and a statistical determination of PL seal performance in three P-40 engines in an automotive-type test cycle. The goal for 1983 is to complete the development of main seals and piston rings through rig and engine testing.

PISTON RINGS

The most promising piston ring design evolved in the ASE Program is the H-ring (shown in Figure 2-27), which is pressure-balanced so that the force holding it in contact with the cylinder wall is independent of the working gas pressure. A separate, internal expander ring is employed to exert a known, constant-radial force on the ring to maintain contact with the cylinder.

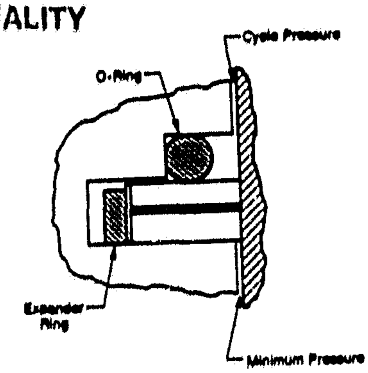

Figure 2-27 Pressure-Balanced H-Ring

By changing the ring dimensions and/or expander ring, designing for a specific, predictable friction level is possible.

Piston rings of the type shown in Figure 2-27 were evaluated in the Exploratory Seals Test Rig (see MTI Report No. 82ASE278SA2). Measurements indicated that friction force was basically independent of gas pressure, as expected, and of the magnitude predicted (see Figure 2-28). Hydrogen leakage showed some variation, but was generally <10 ℓ/min* per ring (Figure 2-29). An analytical study showed that a leak rate of 10 ℓ/min per ring will have a negligible affect on power developed by the engine. If the rings perform as well in the engine under cyclic pressure as the rig tests indicated, they will give a substantial reduction in friction power (shown in Figure 2-30).

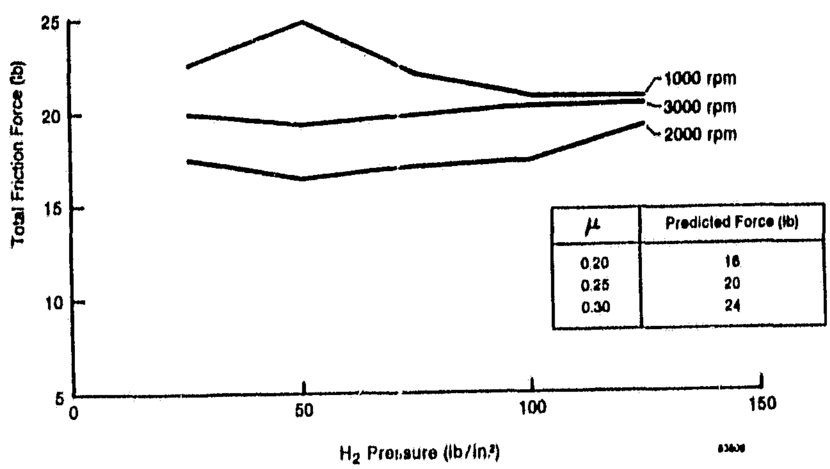

Figure 2-28 Total Friction Force for 4 H-Rings

*approximately an order of magnitude greater than that previously measured with the split/solid-ring design currently employed in the Mod I engine.

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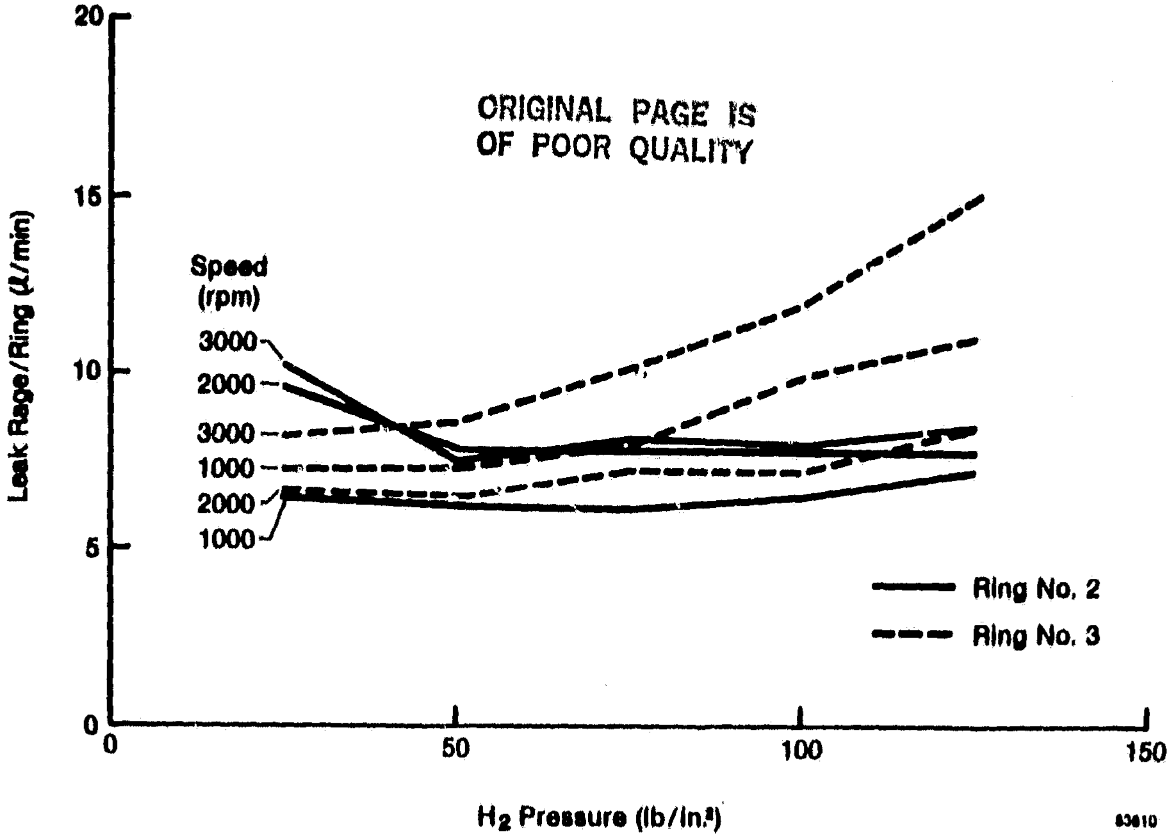

Figure 2-29 Hydrogen Leakage for H-Rings with Butt Joint

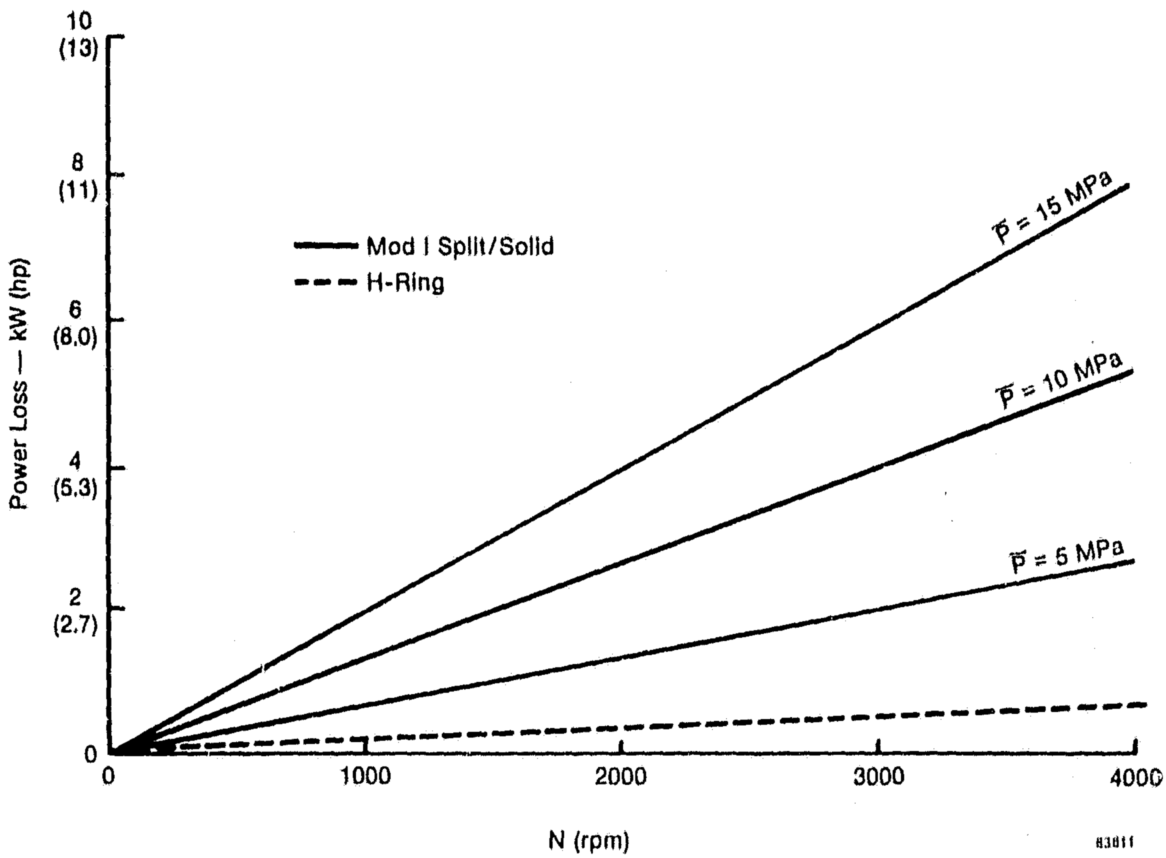

Figure 2-30 Comparison of Total Piston Ring Power Loss
for Mod I Split/Solid Rings and for H-Rings

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Engine hardware was modified to permit further evaluation of the H-ring concept in the motored engine. Double lap-joint-type rings (see Figure 2-27) were installed to reduce gas leakage, but initial tests were complicated and confused by start-up problems with rig hardware. After several unsuccessful attempts to test the rings, it was suspected that gas leakage past the piston ring o-rings (and possible contact between the o-rings and cylinder walls) might be contributing to the problem.

To resolve these questions, the piston ring grooves were remachined, and the ring design was modified as shown in Figure 2-31. This L-shaped ring retains the same pressure-balancing principle of the original design; the added extension of the ring simply acts as a backup ring to prevent contact between the o-ring and cylinder wall if any reversal of pressure across the o-ring occurs during transient operation. Preliminary test data with butt-jointed rings of this form are encouraging, but further testing is needed to fully evaluate its potential. If testing is successful, the piston rings will be installed in a Mod I engine for thorough evaluation.

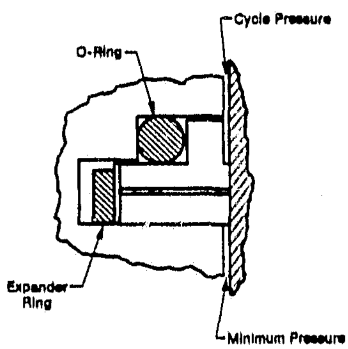

Figure 2-31 Modified H-Ring

MAIN SEALS

The short life experienced with PL seals in the early stages of Mod I testing reconfirmed the requirement for reliable, long-lived rod seals, particularly when a seal failure might allow oil contamination in the engine. The short life of the PL seals in the Mod I engine was inconsistent with USAB's experience with this seal type in engines such as the P-75 and V160.

An examination of good and bad seals gave some indication that variability in seal life may be associated with variations in the properties of the HABIA seal material. To minimize these variations in the future, all main seals will be manufactured from material that meets a known specification. To permit a thorough assessment/analysis of seal performance, each seal will be inspected and serialized before use, a detailed record of its use will be kept, and it will be reinspected after use.

Based on available data, PL seal life appears to be either short or long, with very few intermediate failures. If a seal does not fail within 500 hours, it will probably continue to operate satisfactorily for a much longer period. In order to get a reliable indication of main seal life, three P-40 engines are being operated at USAB primarily for main seal evaluation. This facility will provide a reasonably sound statistical basis for estimating seal life. In P-40 engine No. 4, a set of seals has completed 500 hours with no failures.

During this report period, twelve sets of seals were investigated in the Exploratory Seals Rig (see in Table 2-8).

TABLE 2-8

SEAL SETS INVESTIGATED IN
EXPLORATORY SEALS RIG

Seal Set	Seal Design	Seal Material	Preload (lb)	Test Duration (hr)	Overall Average Hydrogen Leakage (cc/min)		Type of Test
					Top Seal	Bottom Seal	
10	DA $\alpha = 5°$ $\beta = 8°$	Rulon LD	90	140	41	120	C
11	DA $\alpha = 5°$ $\beta = 8°$	Rulon LD	90	123.5	6	27	C
12	DA $\alpha = 5°$ $\beta = 8°$	Rulon LD	40	528	6	3	C
13	PL	HABIA	90	185	23	4	C
14	PL	HABIA	90	187	22	0.6	C
15	DA $\alpha = 5°$ $\beta = 8°$ Relieved	Rulon J	90	26	27	0.6	C
16	DA $\alpha = 5°$ $\beta = 8°$	Rulon LD	40	137	9	0.5	C
17	PL	HABIA	90	60	5.3	1.3	I
18	DA $\alpha = 5°$ $\beta = 30°$	Rulon LD	40	43	3.9	1.1	I
19	DA $\alpha = 5°$ $\beta = 30°$	Rulon LD	40	38.5	0.6	0.7	I
20	DA $\alpha = 5°$ $\beta = 30°$	Rulon LD	40	32	0.5	0.4	I
21	PL	HABIA	90	75	2.1	1.3	I

DA = Double Angle
PL = Pumping Leningrader
α and β defined in Figure 2-32
C = Continuous
I = Intermittent

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In the search for viable, alternative rod seals, the concept of a double-angle seal was generated. With this seal, a thin but finite film of oil separates the rod and seal surfaces while the engine is running. Oil is supplied to the piston rod on the crankcase side of the seal for cooling and lubrication. The oil film in the rod/seal interface is generated by the seal geometry and motion of the rod on the downstroke; oil in the piston rod is returned to the crankcase by similar action. The oil film prevents wear of the seal, and improves the quality of the gas seal. A material with low elastic modulus is preferred for this design, and a static seal is maintained by an interference fit of the seal on the rod.

The main seal test head was designed with conical seal seats (compatible with the engine design at that time) machined into the housings. To investigate the double-angle-seal principle, the seal design had to be capable of fitting into the existing housing. The design adopted is shown in Figure 2-32.

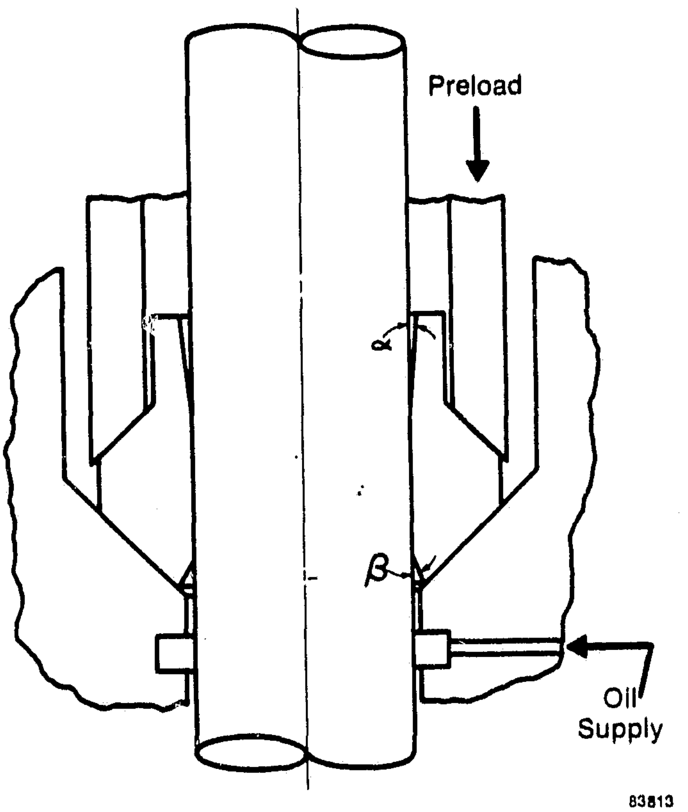

Figure 2-32 Double-Angle Seal in Exploratory Rig

Seal sets 10 and 11 were made from Rulon LD, and had 5° and 8° chamfer angles. With continuous operation, seal set 10 gave low gas leakage and no detectable oil leakage for ~130 hours, after which there was a rapid, progressive increase in gas leakage, resulting in a higher overall average gas leakage. When the seals were removed, a small amount of oil was present in the seal cavity.. On both seals, the chamfer on the low-pressure side had disappeared, leading to the conclusion that this resulted from deformation since there had been no measurable weight reduction in the seals during testing. A repeat test with seal set 11 gave similar results, with the increase in gas leakage taking place after ~115 hours.

To overcome the deformation problem, seal set 12 was installed with a lower preload, 40 lbs instead of the 90 lbs used with seal sets 10 and 11. With continuous testing, seal set 12 ran for 528 hours with consistent low hydrogen leakage (see Figure 2-33). When the seals were removed, no significant deformation had occurred. Oil was present in the seal cavity, but the amount was comparable to that seen in previous tests with PL-type seals after a much shorter time.

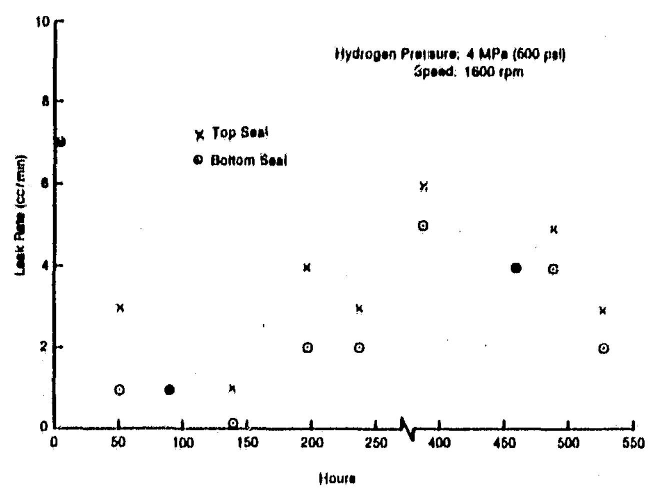

Figure 2-33 Hydrogen Leakage for Double-Angle Seals (Seal Set 12)

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Early rig testing of PL seals was carried out under intermittent conditions. To establish the influence (if any) of continuous operation, a pair of PL seals (seal set 13) were tested under these conditions. The lower seal gave consistently low hydrogen leakage, while the upper seal gave erratic gas leakage throughout. The test was terminated after 185 hours when gas leakage became excessive.

When the seals were removed, only a slight trace of oil was apparent near the seal sets. A repeat test of continuous-running PL seals (seal set 14) gave results very similar to set 13. In these two tests, oil leakage was the lowest of all the seal tests up to that time, and the seals maintained a reasonable gas leakage for a much longer time than the PL seals tested under intermittent conditions. From this, it was concluded that intermittent operation resulted in significantly shorter seal life. Since intermittent operation is more representative of an automotive application, starts and stops will be included in the test cycle for future rig evaluation of seals.

Seal set 15, a slightly modified, double-angle seal design is shown in Figure 2-34. The lower conical surface of the seal was partially relieved in an attempt to reduce the radial load in the 8° chamfer region and prevent deformation. These seals were installed with a 90-lb preload. The bottom seal gave low gas leakage; however, gas leakage from the top seal was high throughout testing. The test was terminated after 26 hours, and the seals removed. Careful examination of the top seal showed that the bore was noncircular, accounting for the high gas leakage. More significantly, the relieved sections of both seal sets had almost disappeared due to deformation of the unrelieved section under the increased contact pressure, obviously making this seal design impractical. As a result, no further tests were conducted.

A repeat test on a double-angle seal wih 40-lb preload (seal set 16) did not reproduce the good performance of seal set 12. The seals were removed after 137 hours, revealing that the 8° chamfers had disappeared because of deformation. Clearly, the reduced preload could not be relied upon to produce consistent seal performance.

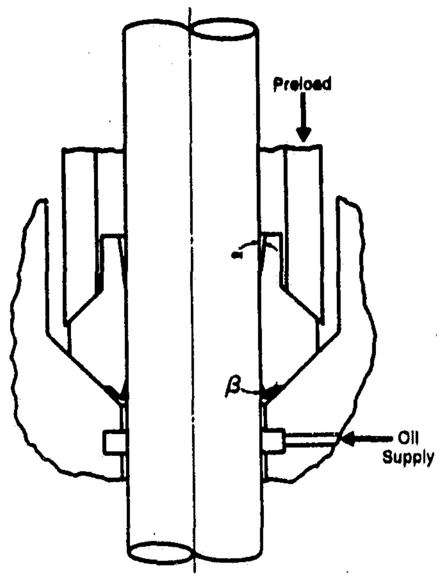

Figure 2-34 Relieved Double-Angle Seal

In the main seal test head, thermocouples in the seal housings close to the seals are used to monitor seal temperatures. In seal set 17, thermocouples were embedded in HABIA PL seals within 1 mm of the seal/rod interface in order to evaluate whether thermocouple temperatures were representative of seal temperatures. Throughout the tests, the thermocouples in the seals indicated slightly higher temperatures than those in the housing, but the temperature difference did not exceed 4°C, leading to the conclusion that housing temperatures were an adequate representation of seal temperatures for normal test purposes.

Seal sets 18, 19, and 20 (all double angle with 5° and 30° chamfers) had 40-lb preload, and were tested intermittently. Testing of seal set 18 was terminated after 43 hours when the temperature of the lower seal became excessive (120°C+). It was found that the 30° chamfer had disappeared from that seal due to deformation. The piston rod was also slightly worn (≈0.001 in.) in the region where the lower seal had been operating. The rod was replaced prior to further testing.

Seal sets 19 and 20 gave low gas leakage, but testing was terminated when a trace of oil was detected in the seal cavity. In seal set 19, the upper seal was dry, but the lower seal had leaked a small amount of oil. The 30° chamfer was still present on the upper seal, but it had disappeared from the lower seal due to deformation. The same occurred with seal set 20.

Testing of double-angle seals has shown that seal type is capable of maintaining low gas leakage for a time and, in the case of seal set 12, for a long period of time. The main indication is that the seals gave low gas leakage and probably negligible oil leakage while seal geometry was retained. In virtually every case, failure was precipitated by deformation of the seal, which destroyed the chamfer on the low-pressure side of the seal. It is reasonable to assume that the deformation was a continuous process that would progressively decrease the angle of the lower chamfer, thus increasing the seal's pumping capacity on the upstroke until it eventually exceeded its pumping capacity on the downstroke, giving a net flow of oil to the gas side of the seal, and constituting a seal failure. Clearly, overcoming the deformation problem was necessary before the double-angle-seal principle could be fully developed.

The dominant force on the seal is due to gas pressure which tends to extrude the seal through the clearance between the rod and seal. In the seal designs tested, this effect would be more pronounced with the unsupported chamfer on the low-pressure side of the seal, which explains the deformation of the seals during operation. One potential method of overcoming the deformation problem is to remove the chamfer from the loaded region of the seal by adding an extension to the low-pressure side. With the integral seal seats in the Exploratory Test Rig, testing of this type of seal was not possible, so a new test bed was designed in which the seals were separate from the main housings (shown in Figure 2-35).

This arrangement is also compatible with the current Mod I/Mod I-A engine hardware, and provides more flexibility for testing different seal geometries without major rig modifications. Apart from this and other detailed changes, the modified test head retains the same overall form as the original head.

The new test head is now available, and baseline testing with HABIA PL seals (seal set 21) has been completed. These seals ran for more than 75 hours under intermittent conditions with low gas leakage. Some oil was present in the seal/seat interfaces, but no gas leakage was evident.

Rig testing during the first half of 1983 will be centered on modified, double-angle seals to evolve a reliable design. When this is achieved, the seals will be tested for durability in a P-40 engine before introducing them into the Mod I/Mod I-A engines.

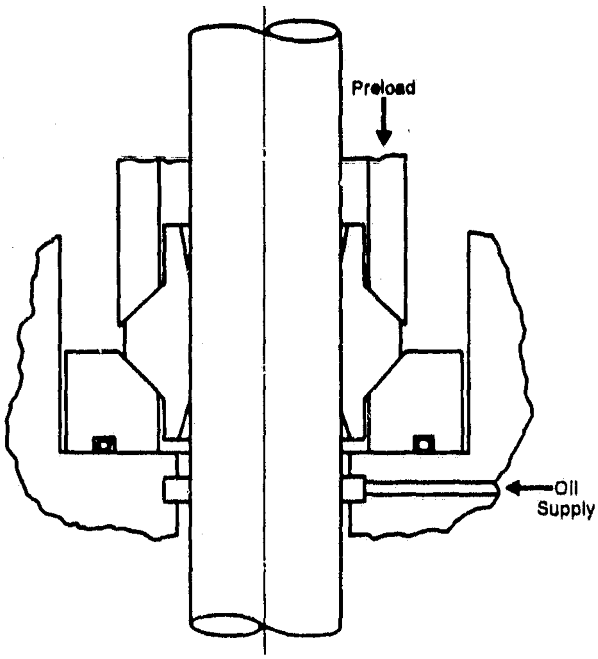

Figure 2-35 Double-Angle Piston Rod Seal

Engine Drive System (EDS) Development

The primary goals of this task are to develop a Reduced Friction Drive (RFD), and evaluate new seal concepts under motored engine test conditions. Activity began in the latter half of 1981 with the installation of a motored Mod I drive system with a dummy heater head.

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Efforts completed during the first half of 1982 included baseline characterization of the EDS motored friction, and design and characterization of cylinder liner P_{min} venting hardware of the RFD.

Efforts completed during this report period include continued testing/evaluation of seal hardware, fabrication and testing of the RFD, and design, fabrication, and testing/evaluation of the Lightweight Reduced Friction Drive (LRFD).

SEAL HARDWARE TESTS

Fabrication of the cylinder liner P_{min} system for support of H-ring testing was completed during this report period. Figure 2-36 shows the general arrangement of the key elements of the system.

Incorporation of the P_{min} hardware into the EDS provided the proper pressure reference for the H-rings. Numerous tests were conducted with various ring configurations with no real success. Mechanical problems with the baseline EDS increased the indicated torque during initial H-ring testing. This situation was resolved when the RFD was substituted for the baseline drive.

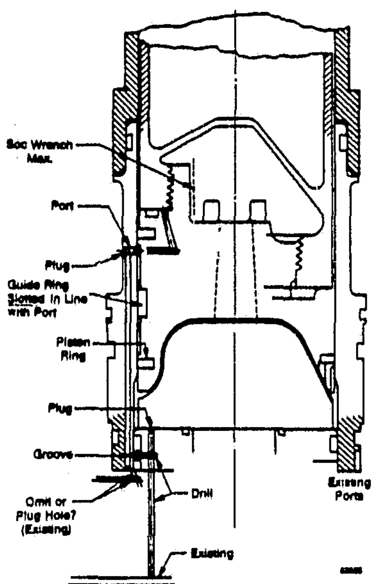

Figure 2-36 General Arrangement of Key Elements of Cylinder Liner P_{min} System

Failure to attain anticipated performance with the H-rings on the RFD prompted an upgrade of the pressure instrumentation in order to obtain more precise cycle pressure data. The gas plumbing was also modified to permit seal housing leakage to bleed back to P_{min}. The latter modification caused the system to maintain better cycle/cycle pressure balance; however, the effects produced by preferential piston seal leakage were masked.

Figure 2-37 shows the four cycle pressures and torque as a function of time for a typical H-ring test with seal housing leakage manifolded to P_{min}; Figure 2-38 shows the same test point with the seal leakage blocked. Note the drastic increase in cycle-to-cycle pressure imbalance, and the resultant torque increase (~100 in. lb.), even though P_{mean} is constant for both tests. It was concluded from testing that the affects of pressure imbalance overpower those of reduced seal friction loading. To provide a more uniform leakage control, seal o-ring and energizing spring revisions will be incorporated into the hardware prior to resuming H-ring testing.

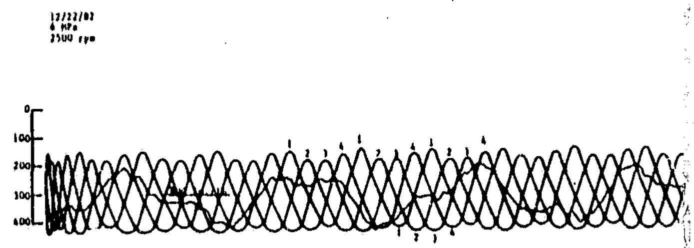

Figure 2-37 H-Ring Test With Seal Housing Leakage Manifolded to P_{min}

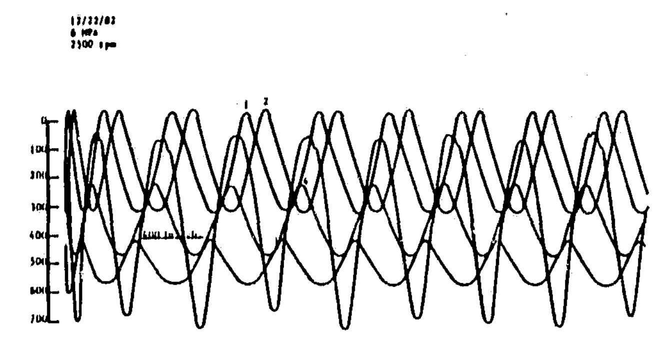

Figure 2-38 H-Ring Test With Seal Housing Leakage Blocked

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REDUCED FRICTION DRIVE

Fabrication and testing of the RFD was completed during the first half of 1982. Figure 2-39 shows the RFD torque versus baseline torque for the drive systems with dummy heads removed, while Figure 2-40 shows the RFD torque versus baseline torque for the complete drive systems with no charge pressure.

Measured torque reduction with the RFD exceeds 2 kW (2.7 hp) at 3000 rpm versus the baseline EDS. The system is a minimal modification of the baseline drive system to permit installation of rolling-element bearings in place of the journal bearings throughout the drive. Figure 2-41 shows the crankshaft/mainshaft bearing details, and the connecting rod bearing details.

The RFD was removed from test in December, 1982 to permit testing of the Lightweight RFD; however, it will go back into test to support seals development in mid-January, 1983.

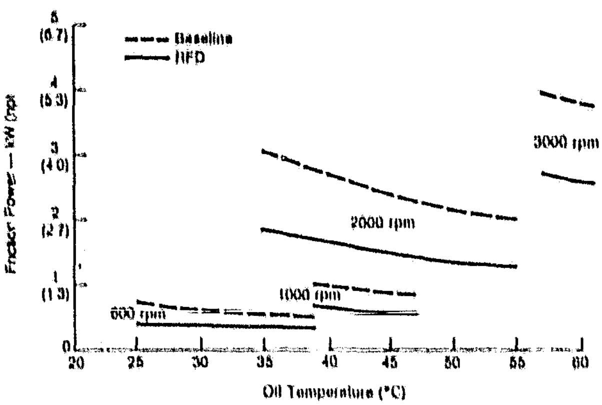

Figure 2-39 Mod I EDS (No Cylinder Liners)

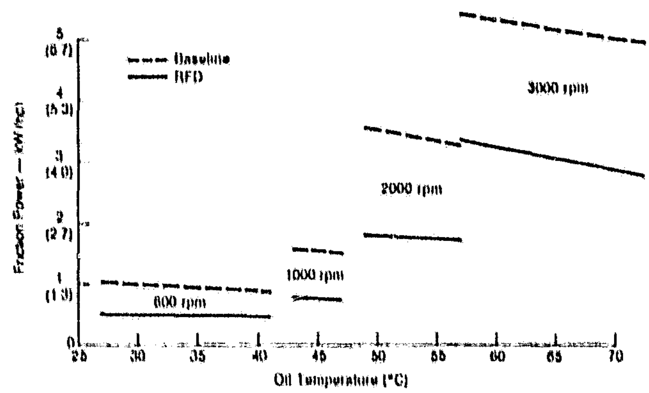

Figure 2-40 Mod I EDS (System Complete - No Working Gas)

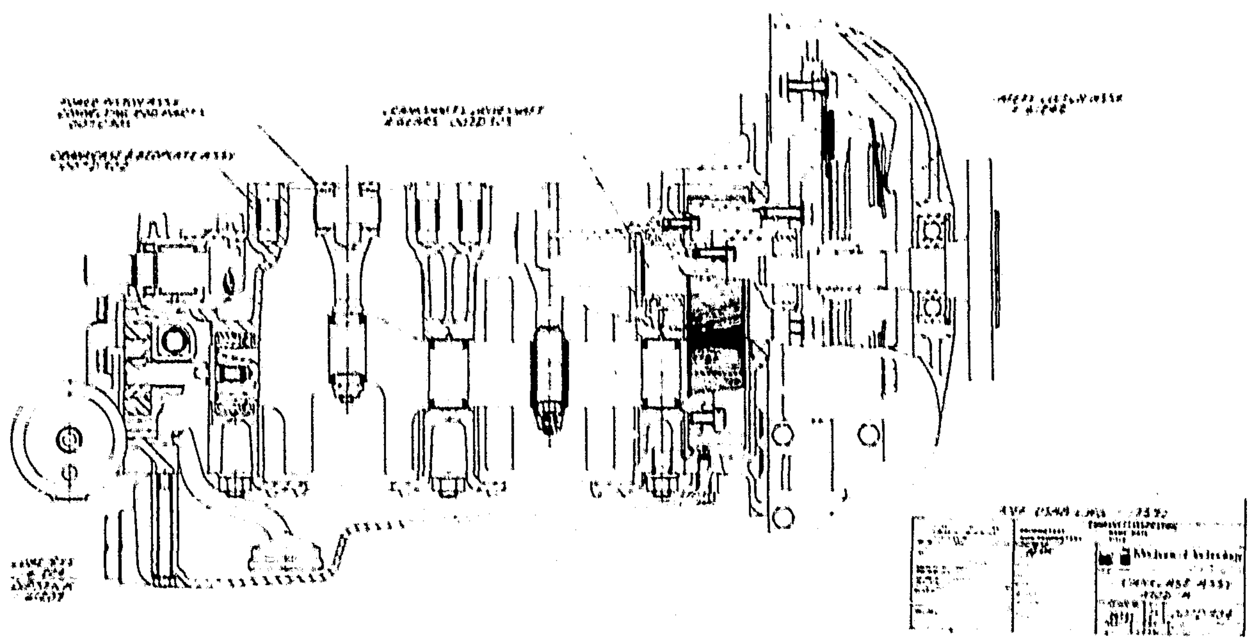

Figure 2-41 Mod I-A Crankcase Assembly

LIGHTWEIGHT REDUCED FRICTION DRIVE

The primary objective of the design, fabrication, and testing of the LRFD, which was completed during this report period, was to combine the performance benefits of the RFD with lightweight hardware designed specifically for application with rolling-element bearings. Reductions of friction power exceeded projections by a substantial margin. Figure 2-42 shows the measured friction power of the LRFD versus the baseline EDS with helium as the working gas, and Figure 2-43 shows comparative data for the two systems with hydrogen as the working gas. The actual weight savings over the baseline EDS is 21 lbs versus a projected 31 lbs. The difference is due to using higher-than-actual baseline crankshaft weights in the original projection, and the decision to use Mod I pistons in lieu of lightweight pistons. This decision resulted in heavier crank counterweights for balance. With testing of the LRFD completed, the unit is scheduled to be built into an engine for the Mod I-A.

Control System Development

The major goals of this task are to develop and evaluate a simple, reliable, driver-acceptable, microprocessor-based electronic control, and an electronic air/fuel control with low pressure drop, low minimum fuel flow, and a programmable air/fuel ratio. These hardware designs must be compatable with the extremes of an automotive operating environment.

Efforts during this report period focused on development and engine-testing of a Repackaged Engine Control/Combustion Control, as well as development of a variable-displacement blower and oil-servo pump.

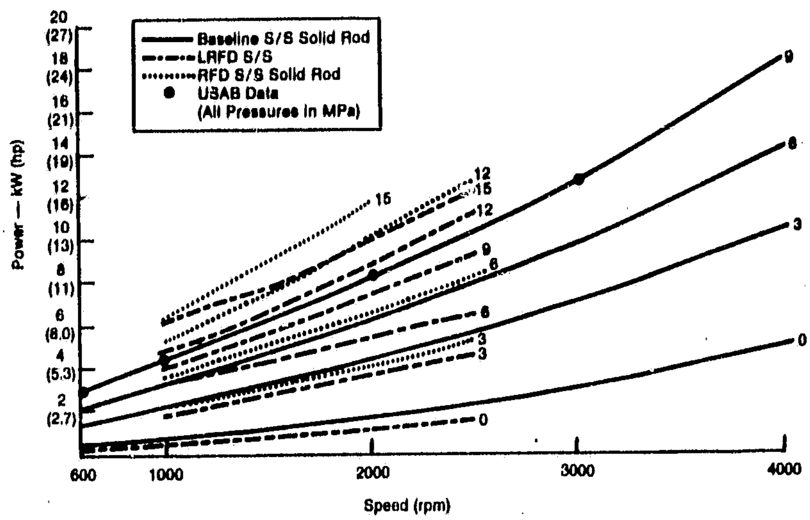

Figure 2-42 Engine Drive System Helium

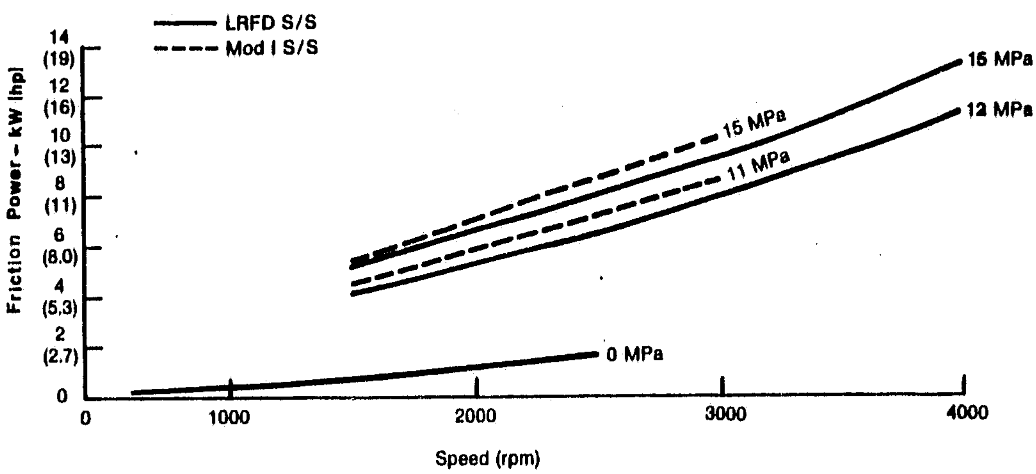

Figure 2-43 Engine Drive System Hydrogen

U.S.A. REPACKAGED ENGINE CONTROL

The objectives of this effort were to enhance the understanding of MTI's Digital Engine Control, produce a package size more suitable for vehicle installation, improve specific design areas, and utilize components readily available through U.S. distributors. The most critical of these objectives was development of a detailed understanding of engine control hardware and software.

The first and, until recently, only Digital Engine Control (prototype configuration only) for the Mod I was designed and built by USAB. The task of reproducing that control was undertaken at MTI to bring knowledge of that control from an informed-user level to a detailed design understanding with capability for better maintaining the system, and the ability to modify the basic control elements when the engine system changes or component testing dictates such a need. Four system design changes were made to enhance the performance of the engine control.

A check of component tolerance stackup indicated that multiplexing of the digital-to-analog control parameter conversions could result in errors. The design was changed to incorporate five separate 8-bit channels and two separate 12-bit channels of digital to analog conversion. The 8-bit channels are used where that conversion resolution is adequate, and the 12-bit channels are used for air throttle control and hydrogen control valve control, where greater conversion accuracy is felt to be necessary. The 8-bit accuracy may be sufficient for those devices, but engine test data is required to make that decision. Solid-state relay drivers were incorporated into the design (instead of locating a relay board within the control module) to increase control reliability and help reduce package size.

The microprocessor chip was upgraded from the IMS 9900 processor to the TMS 9995. They are members of the same Texas Instruments microprocessor family; therefore, the TMS 9995 can execute the same software as the TMS 9900, making it compatible with the original Digital Engine Control software. The TMS 9995 also has additional instruction set features not found in the TMS 9900 such as signed multiply and divide. The execution time of the TMS 9995 is faster than the TMS 9900, providing a potential for shortening the processor cycle time, or addition of control algorithms not currently in the control (i.e., the combustion control) without increasing the cycle time.

A fourth primary modification was made to the software. During its development, some of the original software had become obsolete, replaced by other routines, although never removed from the programming. Other areas of code were needed for testing of the P-40 engine, so the software was changed to eliminate those portions of the code that were not used. Changes were also made to the input/output scaling routines to reduce their execution time. Finally, the Communications Register Unit (CRU) method of handling serial input/output (I/O) (found only with Texas Instruments system) was eliminated. The serial I/O is now done with the widely used method of memory mapping.

Figure 2-44 shows a system diagram of the engine control. The system is made up of four printed circuit boards - Boards 1 and 2 contain the engine control, Board 3 provides power to the engine control boards, and Board 4 contains the relays to drive the final control elements. In a vehicle installation, Boards 1 and 2 would be mounted together under the dashboard, Board 3 would be located near the ignition switch, and Board 4 would be located in the engine compartment.

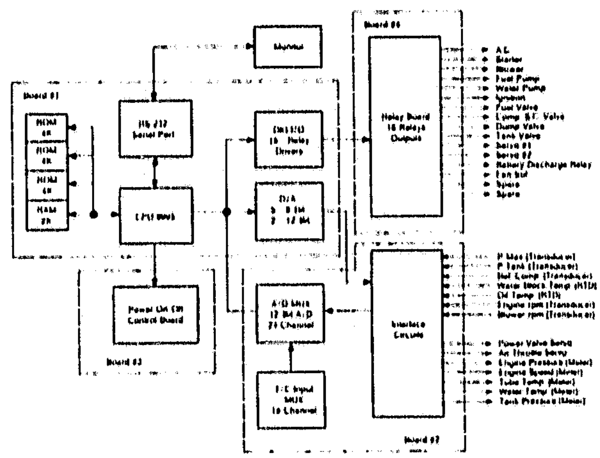

Figure 2-44 System Diagram of Engine Control

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To verify the design modifications to be made, the circuit designs were built and tested, and then integrated and transferred to the system layout described above. The design was then built in a wire-wrapped assembly, maintaining flexibility while the system design was checked out. Based on the wire-wrapped version of the engine control, printed circuit boards were designed and fabricated. In addition, a diagnostic-monitor software package was written to test the engine control hardware functions.

The system design has been proven by its successful operation of a Mod I engine in the engine test cell. The wire-wrapped engine control has been installed in the test cell, providing the engine control since mid-December with no unusual engine-control-related occurrances. A printed-circuit-board version of the engine control has been fabricated and checked out. Installation of that system in the test cell will occur when the test cell schedule permits.

Efforts to reproduce the monitor/simulator were less intense than the engine control effort. The objective was to fabricate a duplicate of the original monitor/simulator within the same package, with cable/communications compatibility to the new engine control, and suitablity for either vehicle or lab installation.

The monitor/simulator is separate from the engine control, and provides an operator interface to the control unit by displaying the engine parameters and permitting simulation of control signals for calibration/system alignment. The monitor/simulator is not required for engine operation, but is crucial to development activity.

Figure 2-45 shows the monitor/simulator system. The major components are the communications processor, the graphics generator processor, the key pad for operator interface, the parameter simulation functions, and the lab panel/CRT display. The lab panel performs the same functions as the vehicle dashboard, and contains the CRT display (would be separate in a vehicle installation).

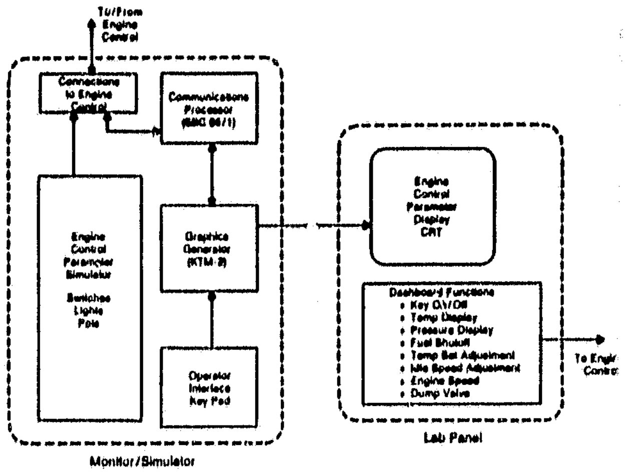

Figure 2-45 Monitor/Simulator System

The monitor/simulator design and fabrication was done in parallel with the engine control work. The two systems were bench-checked together and then installed in the engine test cell. The monitor/simulator has functioned without problem since its installation in mid-December.

Because a design detail understanding of the engine control now resides at MTI, modifications to the control are more easily made. Those changes will be required initially to support the Mod I-A. Several areas have also been identified for changes to control functions and sequencing to improve engine performance. Eventually, the combustion control will be integrated into the engine control electronics package.

COMBUSTION CONTROL

The two objectives of combustion control development are:

- complete the design and evaluation of a Mod I combustion control which improved the performance of the Bosch K-Jetronic system; and,
- the design and operation of a combustion control for the Mod I-A.

The two systems are different because of the fuel nozzles used - the Mod I has a single-stage, air-atomized nozzle, while the Mod I-A uses a two-stage,

pressure-atomized nozzle. To improve combustion control performance, the new control was to provide control down to 0.25 g/s fuel flow, and permit control of λ to any prescribed schedule. Those features are included in both systems.

Rig testing of the Mod I system (see Figure 2-46) has been completed, and the system has been installed in the engine test cell and operated with Mod I engine No. 10. Testing was limited to steady-state operation because of the test cell engine control limitations. The results of that testing were positive.

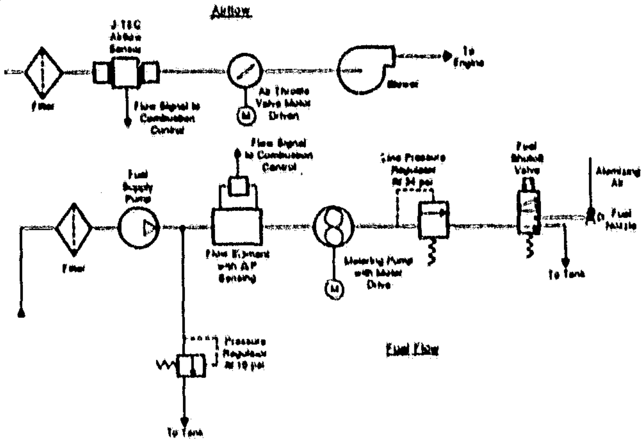

Figure 2-46 Mod I Combustion Control Schematic

The combustion control was successfully operated at fuel flows from 0.25 to 4.0 g/s (minimum design flow to ~90% of maximum flow). Testing demonstrated the functionality of the system, and pointed out some areas where refinements were necessary.

The combustion control operated with greater stability than the K-Jetronic at the 0.25 to 0.50 g/s flow range, as observed by the emissions equipment during Mod I engine testing at MTI. The low fuel flow, exhaust gas oxygen/carbon dioxide concentrations exhibited a better control of λ.

The Mod I combustion control was removed from the test cell and set up on the flow rigs for development of the Mod I-A system (see Figure 2-47).

System development has been completed, and the hardware is ready to be operated with the engine system. Testing with the engine is scheduled for the next available engine test.

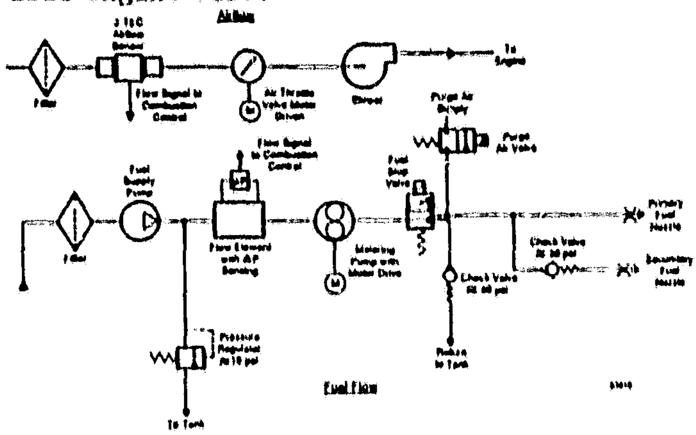

Figure 2-47 Mod I-A Combustion Control Schematic

The combustion control electronics are packaged in a development system that permits flexibility of control algorithm selection, and parameter modification and display. That system is bulky, but a valuable tool for the development process.

Objectives of combustion control development during the first half of 1983 are:

- evaluation with the engine;
- transient testing in the TTB (including emissions);
- integration with the engine control; and,
- the addition of ambient sensing.

AUXILIARIES DEVELOPMENT

A major effort in the area of auxiliaries development during this report period was initiation of a new blower design. This was done to address several problems encountered with the current Mod I blower.

Concern existed over the ability of the current blower to handle EGR flows at high combustor flows, and problems with the blower drive bearings and belt at high operating speeds. With a new design, elimination of the need for the air throttle valve/variator was felt to be possible.

Control of the combustion airflow will be done via a stepper motor, thus eliminating the need for the air throttle valve/variator and variator control, because the blower flow is no longer tied to engine speed. The blower design is complete, and the hardware required to fabricate a prototype blower has been ordered. The prototype will be used to test blower design and performance, and determine the control requirement.

Another major effort in the area of auxiliary development was the servo-oil system, which displayed an inability to develop a design system pressure of 400-600 psi at engine speeds up to 2000 rpm. Analysis and testing of the components has resulted in the following:

- pressure regulator had a steady-state leak rate of 0.58 gpm at 600 psi;
- system was configured to continuously pump against system pressure; and,
- pump performance (speed/flow) appeared to be below specification.

With the regulator set to 600 psi, the pump flow soon went to zero.

Three areas were identified for improving system performance - oil used, housing design, and pressure regulator system. Oil viscosity was increased by conversion from ATF to 10W40. The servo-oil pump housing was redesigned to reduce internal leakage by adding "shadow" parts and changing the inlet porting to reduce inlet velocity. The housing material was changed from aluminum to cast iron due to the influence of differential coefficient of expansion. An unloading valve was adapted so that the system pumps against system pressure only when flow is required.

Figure 2-48 shows the original servo-oil system; the redesigned system, incorporating the new pressure regulation system (modified Citroen system) is shown in Figure 2-49. Testing of the component changes indicated that the unloading valve pressure regulator performed adequately to reduce power consumption and servo-oil heating (Figure 2-50). The pump housing modification did not, however, make the anticipated improvements. Revised calculations for pump performance indicated that the pump was undersized for operation at low engine speeds. A new pump housing has been designed that accommodates a larger pump element within the same outside housing dimensions. (It was necessary to maintain the housing dimensions because of the current mounting space available on the engine.) The pump housing and element for the larger pump have been ordered. Upon receipt, they will be bench-tested and then installed in the Lerma vehicle.

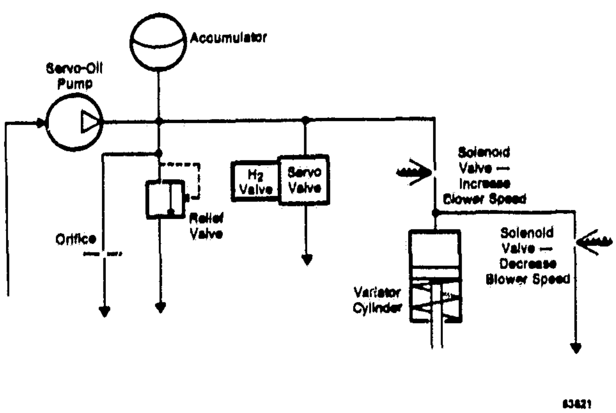

Figure 2-48 Original Mod I Servo-Oil Supply System

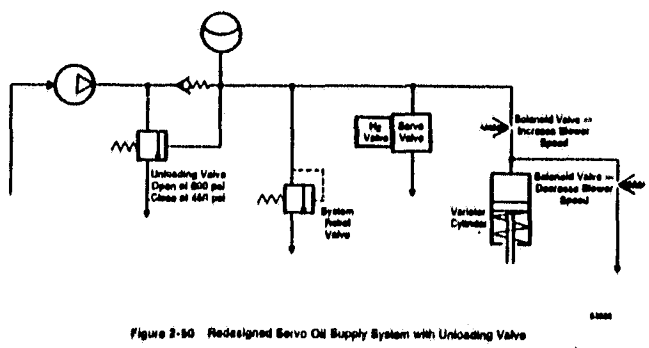

Figure 2-49 Redesigned Servo-Oil Supply System with Unloading Valve

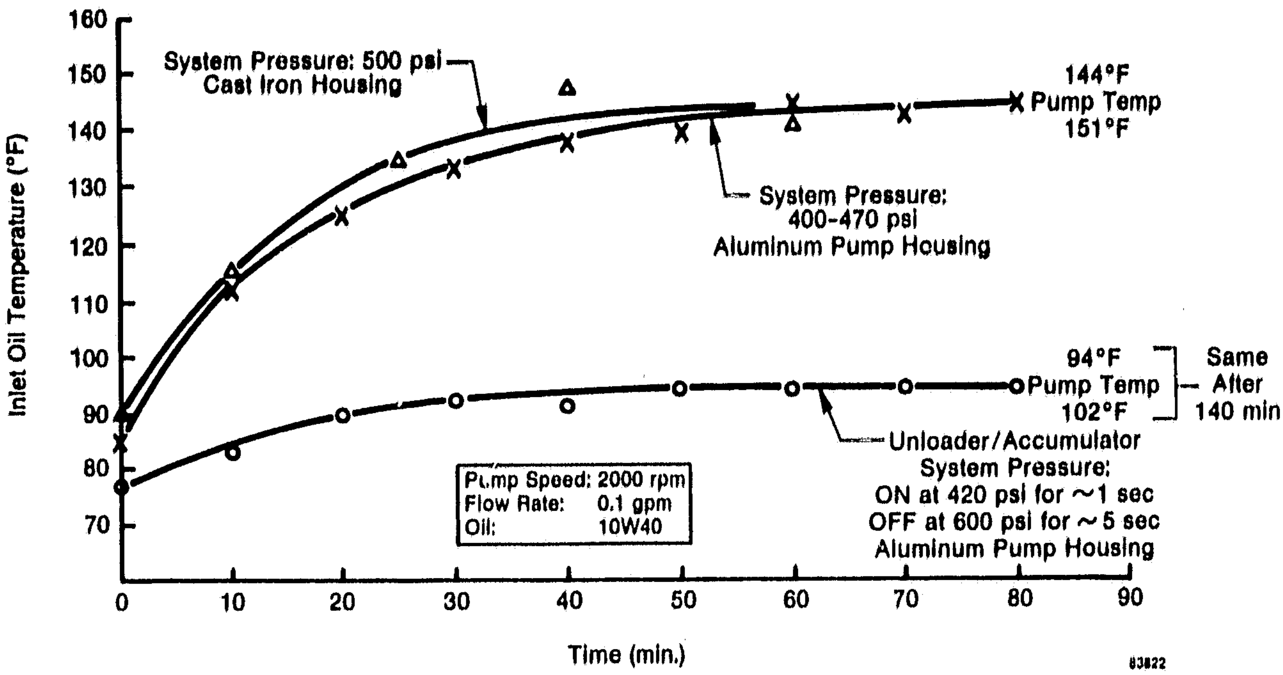

Figure 2-50 Servo-Oil System Test

LERMA VEHICLE INSTALLATION

This task has been directed at supporting the testing program for the Lerma vehicle, and ensuring that the controls are functioning properly.

Most of the electronic controls problems experienced have been corrected by fixing cable connections between the vehicle and the electronic control. An instrumentation panel was installed, providing access to the engine control parameters if a recording device is needed. Those control parameter signals are buffered so the external load will not influence the engine control operation.

The cooling fan logic card was installed and calibrated. The cooling fan is off if coolant temperature is below 35°C, and on if the temperature is above 80°C. Between those two temperatures, the fan is off if vehicle speed is above 20 mph (because of ram air cooling) or when the accelerator is depressed more than 90%. CVS testing was run with the fan logic card in the control system.

Lerma baseline testing showed emissions values to exceed those predicted. The K-Jetronic was found to be holding a richer air/fuel ratio than specified, and occasionally λ would go below 1.0 transiently. Adjustment of the K-Jetronic brought the λ schedule to specification, resulting in an improvement in emissions to predicted levels.

During the second half of 1982, Mod I Stirling engine activity included test cell and TTB evaluation of engines No. 1, 2, 3, and 10. Total accumulated Mod I engine test hours for this semiannual report period are 1700* (includes 121.5 TTB hours).

Highlights for the last six months include the completion of two milestones: 1) TTB characterization of Mod I engine No. 1 in September, 1982 (this was the first transient characterization of a Mod I engine); and, 2) the characterization of Mod I engine No. 10 (significant because this engine was manufactured for the most part in the United States). Previous engines evaluated were manufactured and first assembled at USAB. Other developmental work was directed at acquiring more information to aid in the upgraded Mod I engine program. Efforts to understand performance differences between engine EGR/CGR combustion systems, and BSE/SES performance comparisons also took place.

Mod I Engine No. 1

In July, 1982, the TTB engine was converted from a CGR to an EGR combustion system. Since this was a major configuration change, and could result in performance changes, the engine was removed from the TTB and installed in MTI's engine test cell. A complete map of engine performance was then generated with the EGR system (see Figures 3-1 and 3-2). With the new performance maps generated while the TTB engine was in the test cell, data could be used as input into vehicle performance programs, and transient data for engine No. 1 could be predicted.

In September, 1982, the engine was reinstalled in the TTB for completion of its baseline characterization, which consisted of CVS testing (3,750-lb. inertia weight/11.1-hp road load setting) with gasoline. The data acquired, shown in Table 3-1, is cold-start data from an average of three runs on three different days. Emissions in g/mi and urban, highway, and combined mileages are also included in the table.

Mod I engine No. 1 is currently installed in the TTB. Development efforts for the first half of 1983 will be directed toward improving transient engine system performance.

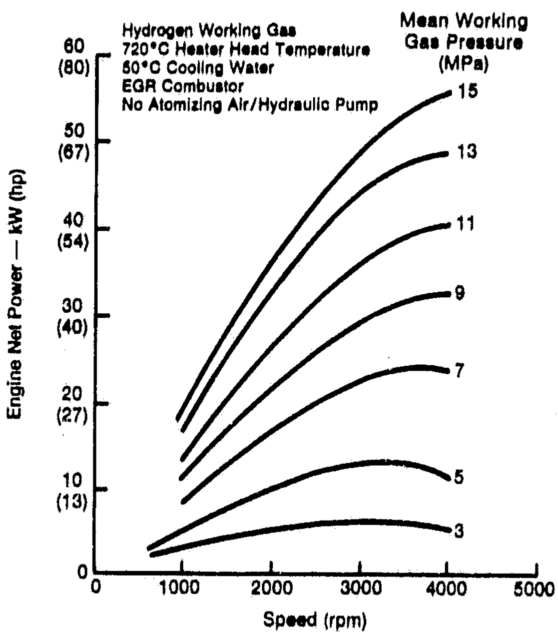

Figure 3-1 ASE 4-123-01, Mod I Stirling Engine System Power

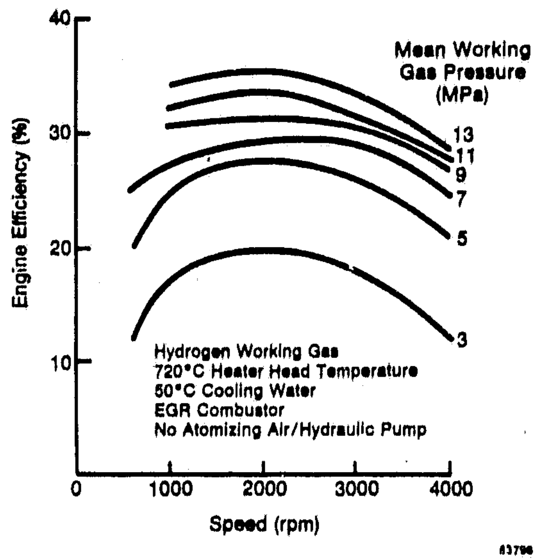

Figure 3-2 ASE 4-123-01, Mod I Stirling Engine System Efficiency

*699 additional hours since last semiannual report period

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TABLE 3-1

LERMA TTB, CVS-CYCLE EMISSIONS/MILEAGE

Inertia Weight	= 3850 lb
Road Load	= 11.1 hp
Fuel	= Indolene
Combustion System	= EGR
EGR Schedule	= Fixed Sized Orifice 23% EGR @ 1.0 g/s Fuel Flow Input

	Urban (g/mi)	Highway (g/mi)
HC	0.254	0.004
CO	3.300	0.310
NO_x	0.900	0.660
mpg	19.30	32.10
Combined mpg	23.50	

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Mod I Engine No. 2

Mod I engine No. 2 was characterized as an SES in October, 1982. The power and efficiency data from this test (shown in Figures 3-3 and 3-4) will be analyzed and compared to BSE data (obtained during the first half of 1982) in the Mod I Performance Analysis Section of this semiannual report.

Following BSE characterization, the engine's drive unit was replaced with another unit in which six reduced size bearing journals had been incorporated. As shown during motoring tests with the reduced size unit, no difference in performance was noted on the engine when run with this unit.

During the first half of 1983, Mod I engine No. 2 will be converted to a Mod I-A engine following oil pump/oil grade testing.

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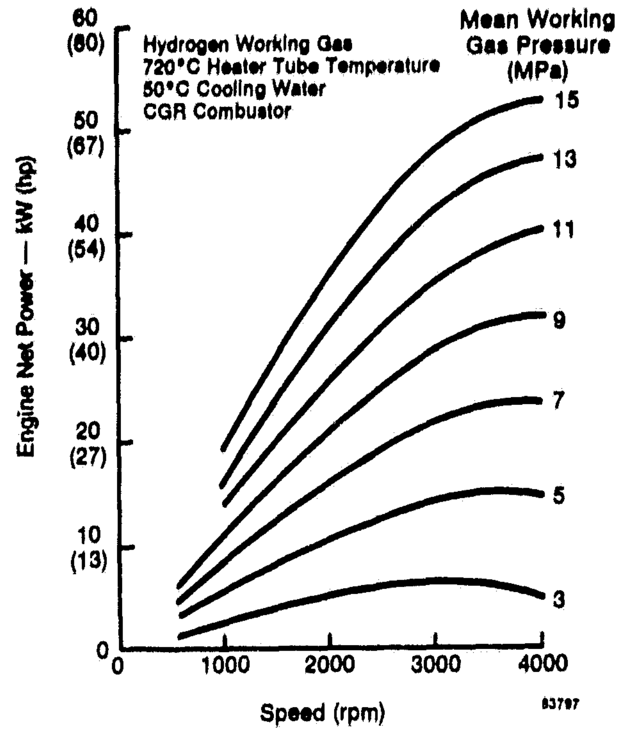

Figure 3-3 ASE 4-123-02, Mod I Stirling Engine System Power

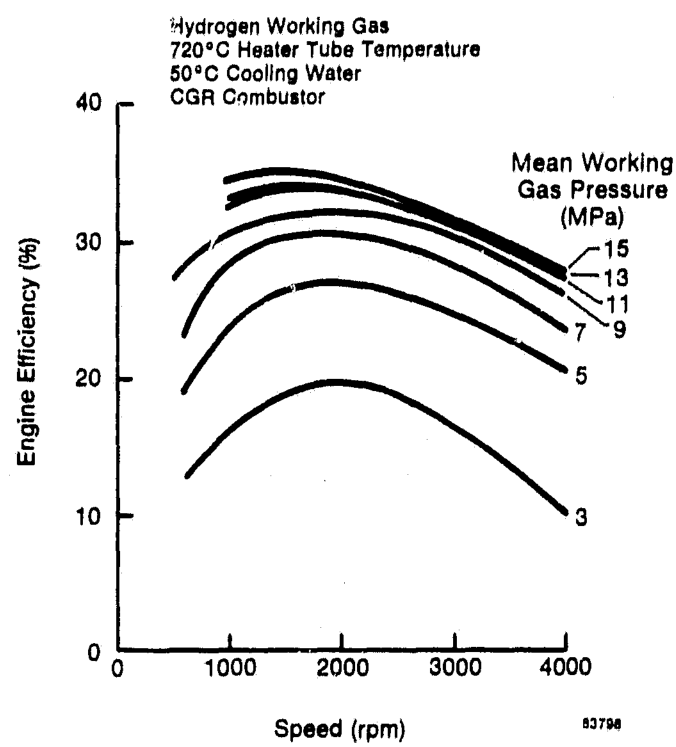

Figure 3-4 ASE 4-123-02, Mod I Stirling Engine System Efficiency

Mod I Engine No. 3

Engine No. 3 was torn down and inspected during the early part of this semiannual report period following completion of its characterization testing at MTI.

In August, 1982, Mod I engine No. 3 was shipped to USAB, where it is currently being prepared to run a 2000-hour endurance test (including 200 starts and stops) that is scheduled to begin early in 1983.

Mod I Engine No. 10

U.S.A.-manufactured engine No. 10 was installed in the MTI test cell in August, 1982, and its characterization testing was completed in early November (power and efficiency data are plotted in Figures 3-5 and 3-6).

Following the engine's characterization, checkouts were made of both the MTI Digital Engine Control and Air/Fuel Control, which will be used on the Mod I-A engine in early 1983. Mod I engine No. 10 was also used to evaluate performance differences between the EGR and CGR combustion systems*.

The engine will be left in the test cell until mid-January, 1983, where evaluations will be made of knitted wire regenerators and an air-blast fuel nozzle (to be used on the Mod I-A engine). Upon completion of this testing, engine No. 10 will be removed from the cell and converted to an upgraded design.

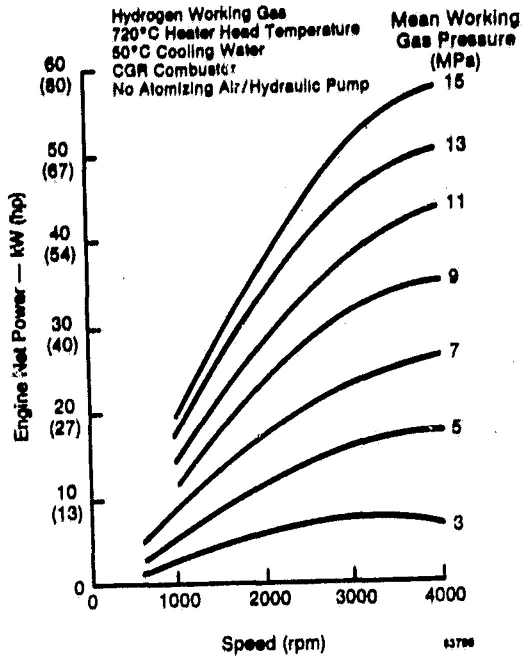

Figure 3-5 ASE 4-123-10, Mod I Stirling Engine System Power

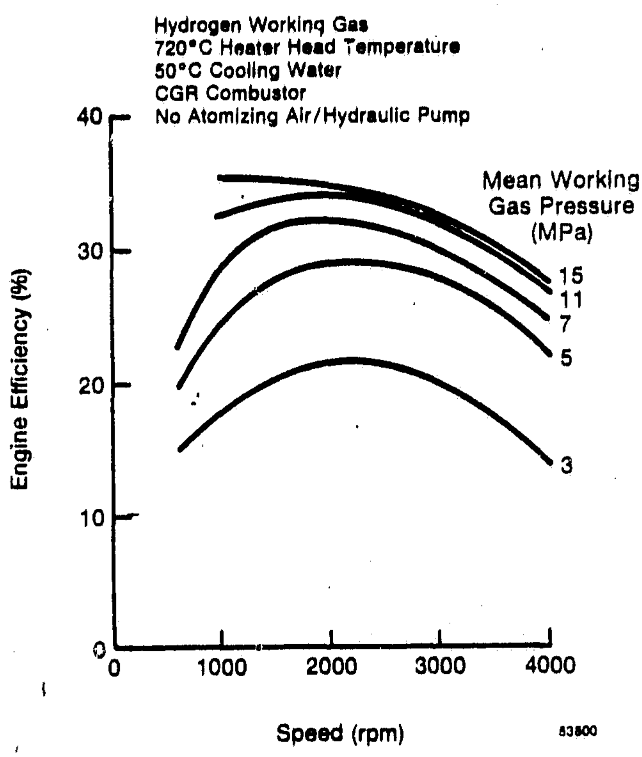

Figure 3-6 ASE 4-123-10, Mod I Stirling Engine System Efficiency

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*see the Performance Analysis Section of this report

Mod I Engine Operational History

Mod I engine testing since the inception of the ASE Program was reviewed to determine utilization rate, and identify problem areas. Figure 3-7 is a chart of the utilization rate since the start of the Mod I program. Included is the utilization rate achieved with ASE P-40-7 at MTI during 1981. It is noteworthy that the average utilization rate for the Mod I engine, even during the first two years of operation, is 50% higher than that achieved on the P-40 engine, a much more mature engine. Mod I design changes focused on reliability have indeed decreased engine downtime.

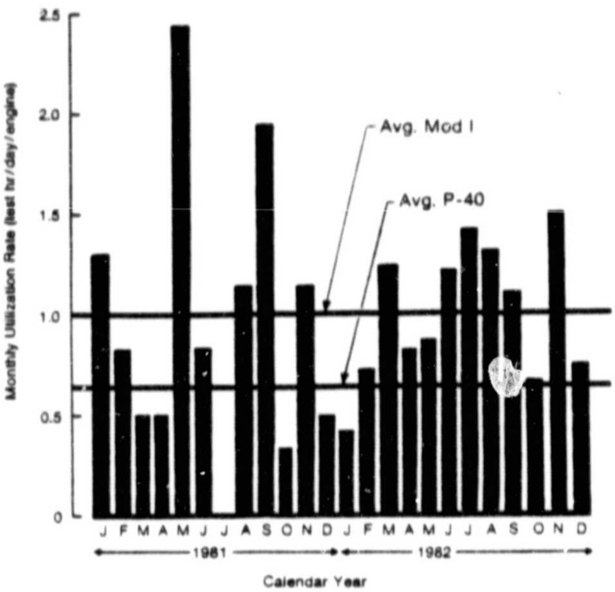

Figure 3-7 Mod I Monthly Utilization Rate

Quality Assurance Reports (QAR's) were reviewed to categorize engine failures or discrepancies, and identify problem areas. Since the objective of this effort is to identify problems that affect engine operation, and also to provide a guide for engine development efforts, any event that has:

- caused an engine to stop running;
- resulted in an engine not starting; or,
- had a detrimental affect on engine operation

was classified as the occurance of a problem. Major individual items and the number of occurrences contributing to these problems are*:

• control thermocouples	40
• MOOG power-control-valve actuator	17
• heater head	8
• engine check valves	6
• combustion air blower	6
• fuel nozzle	6
• igniter	5

The Mod I/Mod I-A development programs, and the RESD design address these items to either eliminate the problem areas or modify the design.

Performance Analysis

LERMA TTB MILEAGE

Mileage obtained in the TTB is compared to that of previous Stirling-powered vehicles in Figure 3-8. A direct comparison of measured fuel economy is not valid due to the vehicles' differences in test weights and power-to-weight ratios.

The P-40 vehicles have been accordingly adjusted in Figure 3-8 to show performance that would result at the same power-to-weight ratios and test weights. As adjusted, Mod I baseline testing represents a 50%/26% mileage improvement over the P-40 Opel/Spirit vehicles, respectively.

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*This data is summarized by engine system in Section V of this report.

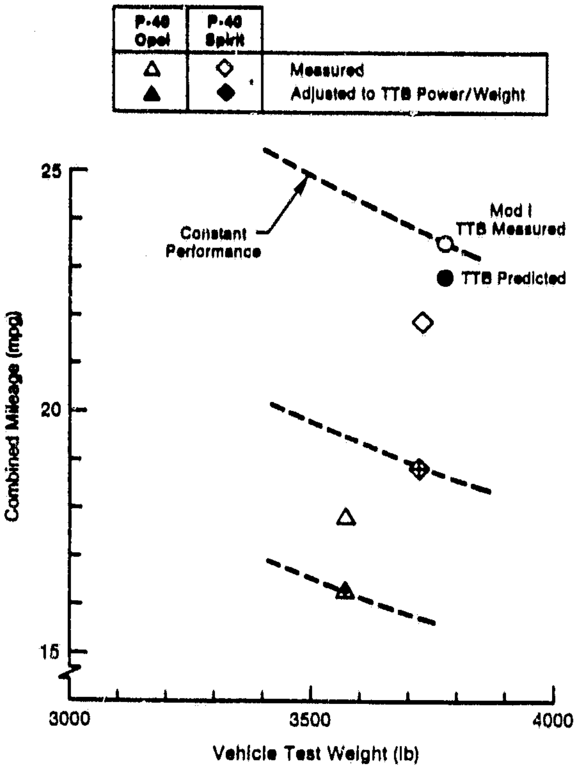

Figure 3-8 Vehicle Fuel Economy Comparison

Figure 3-9 is a breakdown of the energy consumed during the urban/highway cycles for the Mod I Lerma system. As illustrated, significant amounts of total engine energy use (30% urban/38% highway) are consumed by bottom-end engine friction*, auxiliaries, and cold-start penalty.

FINAL ACCEPTANCE TEST DATA REVIEW

A review of final acceptance test data was completed for Mod I engines No. 2, 3, and 10 during this semiannual report period. A comparison of the performance of those engines relative to established criteria is presented in Table 3-2.

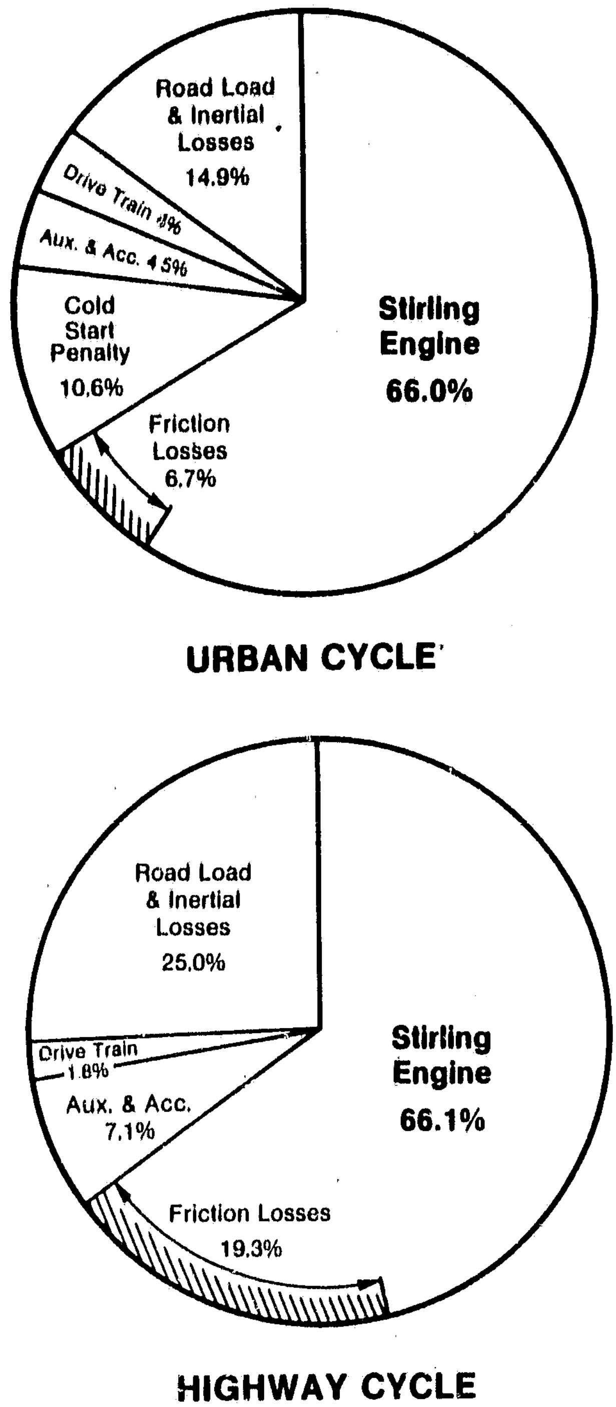

Figure 3-9 Lerma Vehicle Energy Consumption

*drive unit, seals, rings

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TABLE 3-2

MOD I ENGINES FINAL ACCEPTANCE TESTING

Mod I Engine No. 2 BSE Final Acceptance Performance Comparison

Parameter		Condition (MPa/rpm)	Minimum	Actual
Power	kW (hp)	15/1000	19 (25)	20.12 (26.98)
		15/2000	35 (47)	38.09 (51.08)
		15/3000	47 (63)	51.56 (69.14)
		15/4000	53 (71)	58.65 (78.65)
Efficiency	%	15/1500	35	37.27
		5/2000	28	29.92
Emissions		Not Measured		

Mod I Engine No. 3 Partial SES Final Acceptance Performance Comparison

Parameter		Condition (MPa/rpm)	SES	Minimum Partial SES	Actual
Power	kW (hp)	15/1000	18 (24)	18.62 (24.97)	19.20 (25.75)
		15/2000	33 (44)	34.05 (45.66)	36.90 (49.48)*
		15/3000	44 (59)	45.42 (60.91)	49.85 (66.85)*
		15/4000	50 (67)	51.74 (69.38)	56.70 (76.03)
Efficiency	%	15/1500	33	34.10	34.55
		5/2000	25	27.60	28.92
Emissions	g/kg fuel	EIHC	< 1.9		~ 0.2
		EICO	<16.0		~ 3.0
		$EINO_x$	< 2.9**		~ 7.0
			< 7.3***		

*No EGR **0.4 g/mi ***1.0 g/mi

Mod I Engine No. 10 Final Acceptance Performance Comparison

Parameter		Condition (MPa/rpm)	SES	Minimum Partial SES	Actual
Power	kW (hp)	15/1000	18 (24)	18.6 (24.9)	20.4 (27.3)
		15/2000	33 (44)	34.1 (45.7)	38.6 (51.8)
		15/3000	44 (59)	45.4 (60.9)	51.2 (68.7)
		15/4000	50 (67)	51.7 (69.3)	56.9 (76.3)
Efficiency	%	15/1500	33	34.10	35.68
		5/2000	25	27.60	28.68

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Engine No. 2 was tested as a BSE, while engines No. 3 and 10 were tested as complete SES's. Note that for all engines, the established criteria for power and efficiency were met or exceeded at all conditions by all three engines. A total of four Mod I engines have been successfully acceptance-tested to date in the ASE Program.

MOD I ENGINE AUXILIARY POWER CONSUMPTION

Engine testing was conducted to assess the total power consumed by the engine auxiliaries. Engine No. 2 was tested as both a BSE and an SES to determine auxiliary power (the SES configuration includes all engine auxiliaries driven by the engine, whereas the auxiliaries in the BSE are all remotely powered).

Figure 3-10 shows the differences in measured output power between the BSE and SES configurations compared to the summation of power required for the individual auxiliaries as measured by component rig testing. Excellent agreement with rig testing was achieved, with a difference of ~0.7 kW (~0.94 hp) at maximum power.

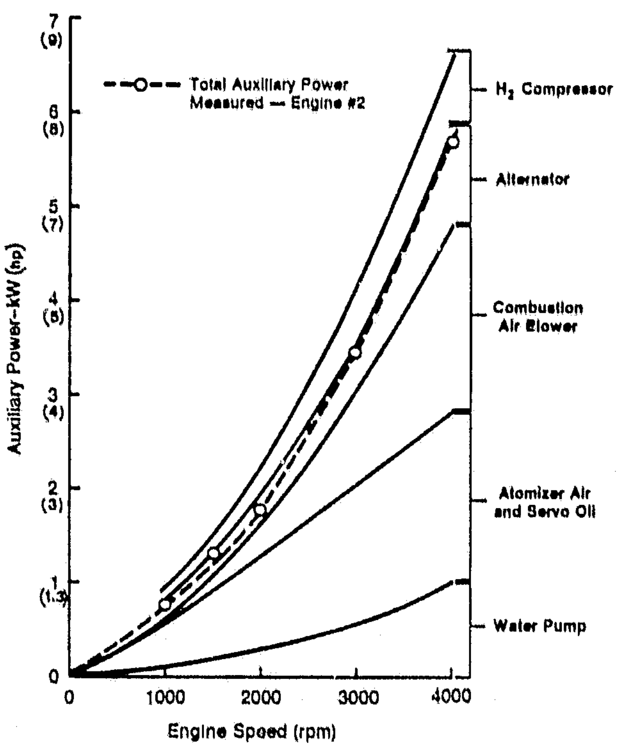

Figure 3-10 Mod I Engine Auxiliary Power (15 MPa)

EGR/CGR ENGINE COMBUSTOR COMPARISONS

Back-to-back tests were conducted during this semiannual report period to evaluate the performance of the Mod I engine with two different combustors - EGR and CGR. EGR-combustor tests were run with and without the EGR activated. Testing results indicated a slight improvement in engine performance with the CGR system. The engine performance curves for net power and efficiency are shown in Figures 3-11 and 3-12, respectively. Lines drawn through the EGR (on and off) data clearly show the difference in performance when EGR is activated. A line has not been drawn through the CGR data, but it is evident that CGR performance is equal to or better than the EGR (on) performance.

The performance difference is attributable to the combination of combustion air blower power (a function of EHS pressure drop and combustion gas flow) and EHS efficiency (ηB). Combustion air blower power for the three configurations is shown in Figure 3-13. Note that the CGR system has the highest blower power requirement due to the increased EHS pressure drop. EHS efficiency is depicted in Figure 3-14. The CGR ηB is equal to or higher than EGR-on ηB. The blower power and ηB differences are small, and are in the direction to confirm measured power and efficiency.

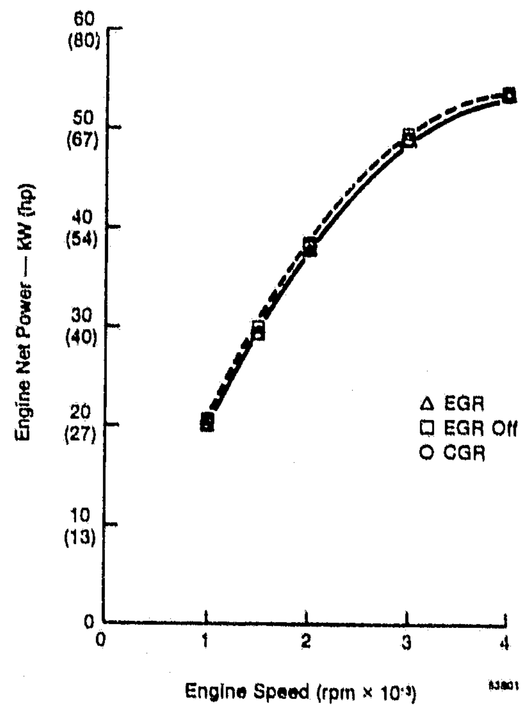

Figure 3-11 Mod I Engine No. 10 (15 MPa) Net Power Versus Speed

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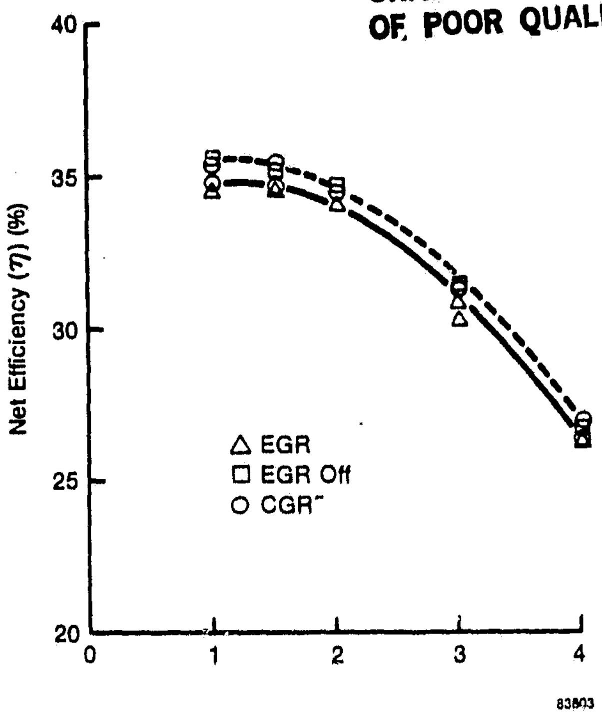

Figure 3-12 Mod I Engine No. 10 (15 MPa) Net Efficiency Versus Speed

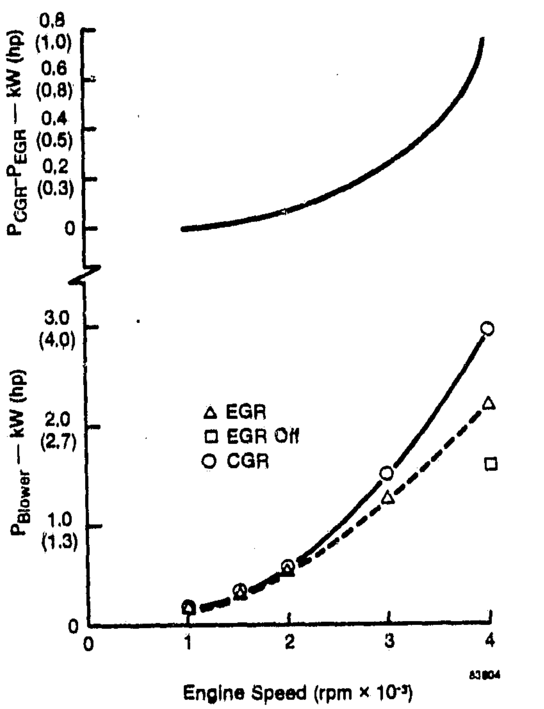

Figure 3-13 Mod I Engine No. 10 (15 MPa) Blower Power Consumption

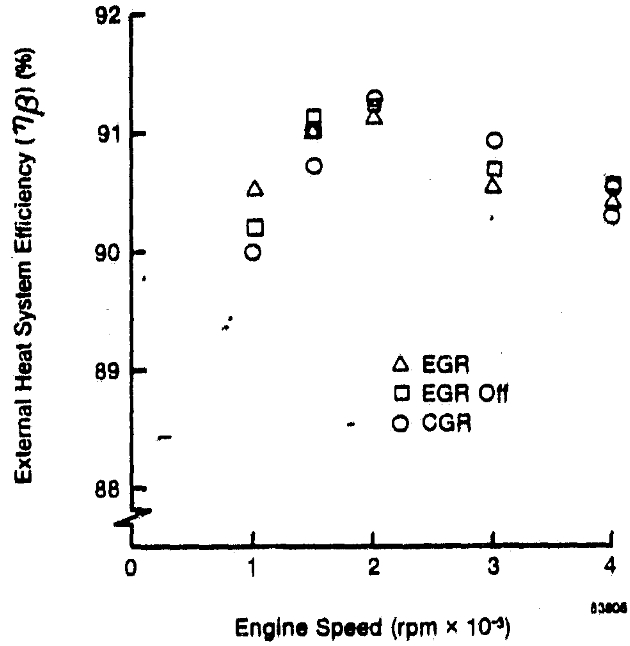

Figure 3-14 Mod I Engine No. 10 (15 MPa) EHS Efficiency Versus Speed

MOD I ENGINE PERFORMANCE COMPARISON

Several different tests have been run on the four Mod I engines currently in the ASE Program. There are many configurational differences in the tests, as shown in Table 3-3. To facilitate an engine-to-engine comparison, performance levels were corrected to a common base configuration, as indicated in Table 3-3. When the corrections are applied to all data, good agreement is achieved on an engine-to-engine basis, with only a few exceptions. The power and efficiency comparison at maximum engine pressure is shown in Figures 3-15 and 3-16. On a power basis, engine No. 10 exhibits a higher power level than the other Mod I engines. The difference is thought to be primarily due to slightly different engine buildup (stainless-steel cylinder liners and perhaps a more carefully controlled build procedure).

On an efficiency basis, the engines all agree within 2%, with the exception of engine No. 1, which was tested at USAB. Configurational differences (high ΔT with a CGR combustor, slightly different working gas drillings, and warped regenerator spacer rings) are probable reasons for engine No. 1's different efficiency level.

TABLE 3-3

MOD I ENGINE DATA COMPARISON CONFIGURATION SUMMARY

Engine/Test	No. 1 Final Accept.	No. 1 Vehicle Base	No. 2 BSE Accept.	No. 2 SES	No. 3 Final Accept.	No. 10 Final Accept.
Location (Stand H_2O Pump)	USAB	MTI	USAB	USAB	MTI	MTI
H_2O Pump Gearing	Lo-Speed	Hi-Speed		Lo-Speed	Lo-Speed	Hi-Speed
H_2 Compressor	Yes	Yes	No	Yes	Yes	Yes
Servo-Oil/Atomizing-Air	Yes	No	No	Yes	No	No
Combustor	CGR	EGR	CGR	CGR	EGR	CGR
Alternator	No Load	Electronics	No Alt.	Electronics	Electronics	Electronics

////// = base

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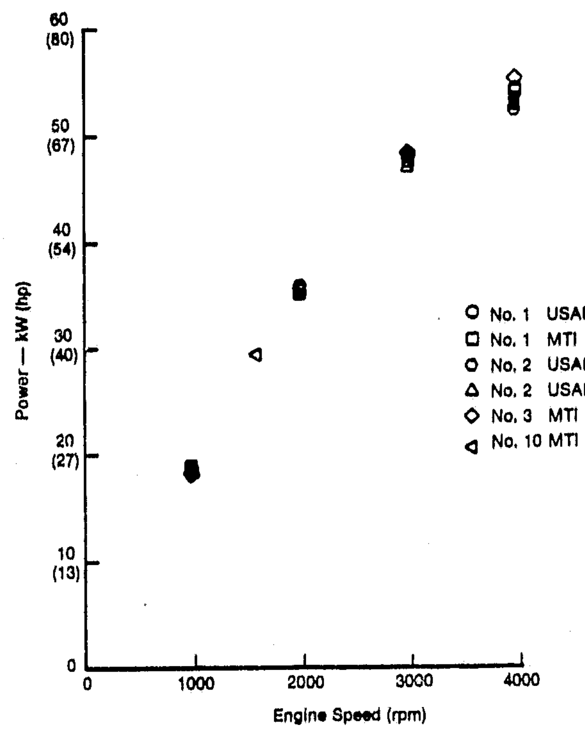

Figure 3-15 Mod I Engine Comparison (15 MPa) Corrected Power

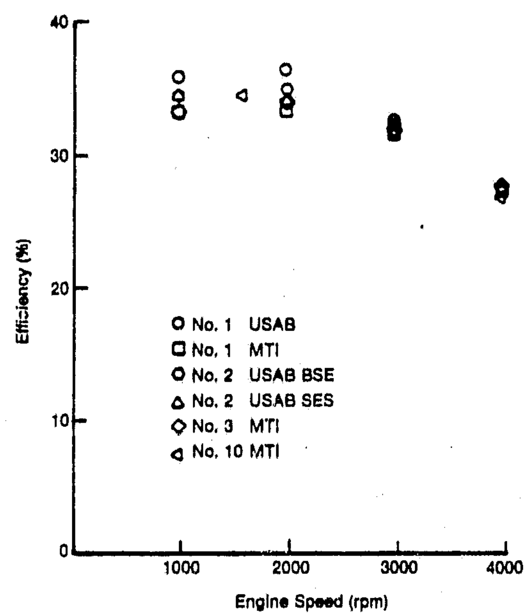

Figure 3-16 Mod I Engine Comparison (15 MPa) Corrected Efficiency

MOD I-A Design

All designs for the Mod I-A engine build were completed during the latter half of 1982. (Mod I-A goals and design concepts were presented in MTI Report No. 82ASE278SA2.) To facilitate early testing of the Mod I-A to characterize performance improvements, certain subsystems of the initial Mod I-A engines will differ from the final Mod I-A design, as noted in Table 3-4. Performance projections for the Mod I-A have been updated, and are presented in Table 3-5.

Mod I-A Hardware Status

A schedule depicting the Mod I-A hardware status is presented in Figures 3-17 through 3-23.

TABLE 3-4

MOD I-A INITIAL CONFIGURATION SUMMARY

System	Component	Final	Initial
External Heat System	Combustor	Delavan CGR	#2-A Has EGR Mod I Nozzle
Hot Engine System	Heater Head	CG-27 Tubes	Inconel 625
Cold Engine System	Piston/Venting	Lighter Piston Liner Venting	Mod I
	Seal Housing	One Piece	Mod I
Drive Unit	Bearing System	Rolling Element	#2-A Has Reduced Size Journal-Type
Controls	Electronic Control Package	Digital Electronic Control	#2-A Has Mod I Control

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TABLE 3-5

MOD I-A PERFORMANCE PROJECTION/MAJOR IMPROVEMENT AREAS

Operating Condition	Parameter	Mod I-A	Δ from Mod I
15 MPa 4000 rpm (Maximum Power)	Net Power Net Efficiency	60.26 kW 29.9%	+10.2 kW +2.7 pt
15 MPa 1500 rpm (Maximum Efficiency)	Net Efficiency	40.76%	+3.07 pt
5 MPa 2000 rpm (Average Operating Point)	Net Power Net Efficiency	12.57 kW 33.04%	+2.43 kW +4.95 pt
	Specific Weight kg/kW lb/hp	 4.4 7.3	 -24%

Change	ΔMaximum Net Power	ΔMaximum Net η
Increased Set Temperature (720° → 820°C)	+7	+1.0% pt
Part Power Optimization	-2	+1.0% pt
Decreased Friction Losses	+1.2	+0.7% pt
Elimination of Atomizer Air Compressor	+0.5	+0.3% pt

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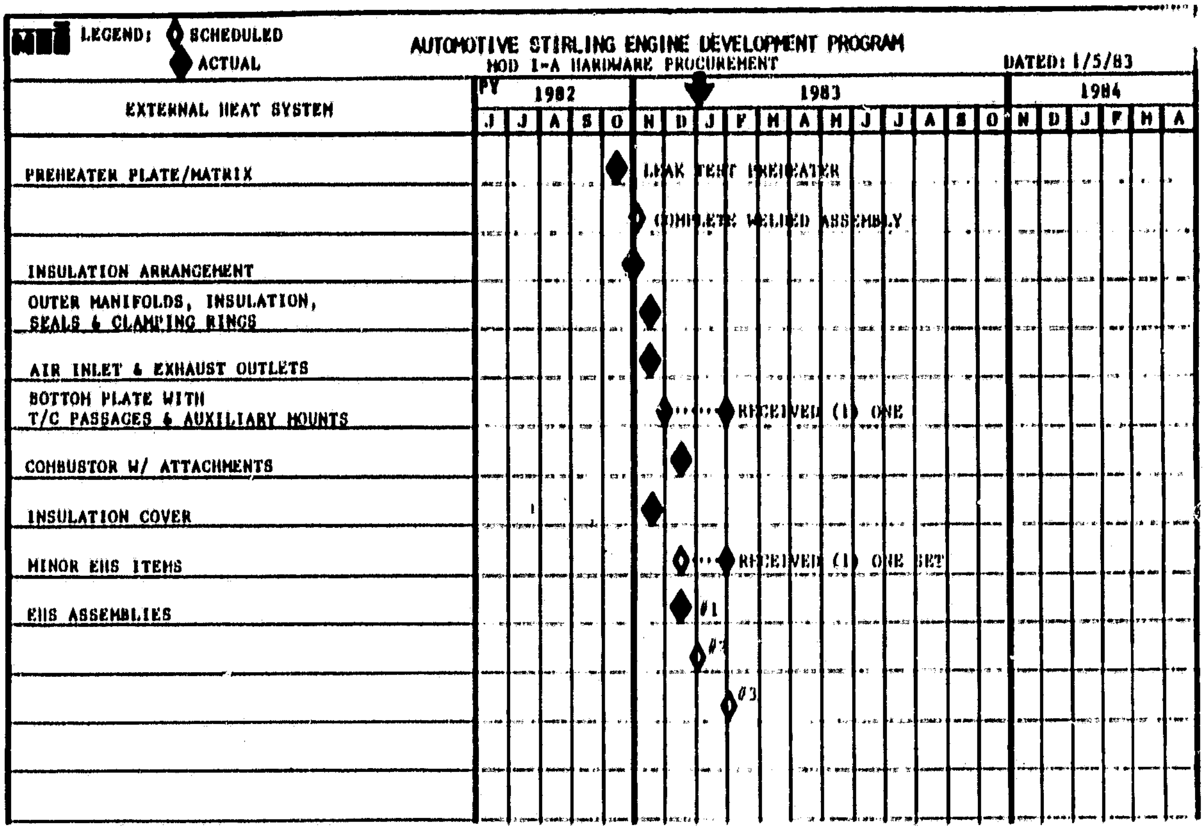

Figure 3-17 Mod I-A Hardware Procurement - External Heat System

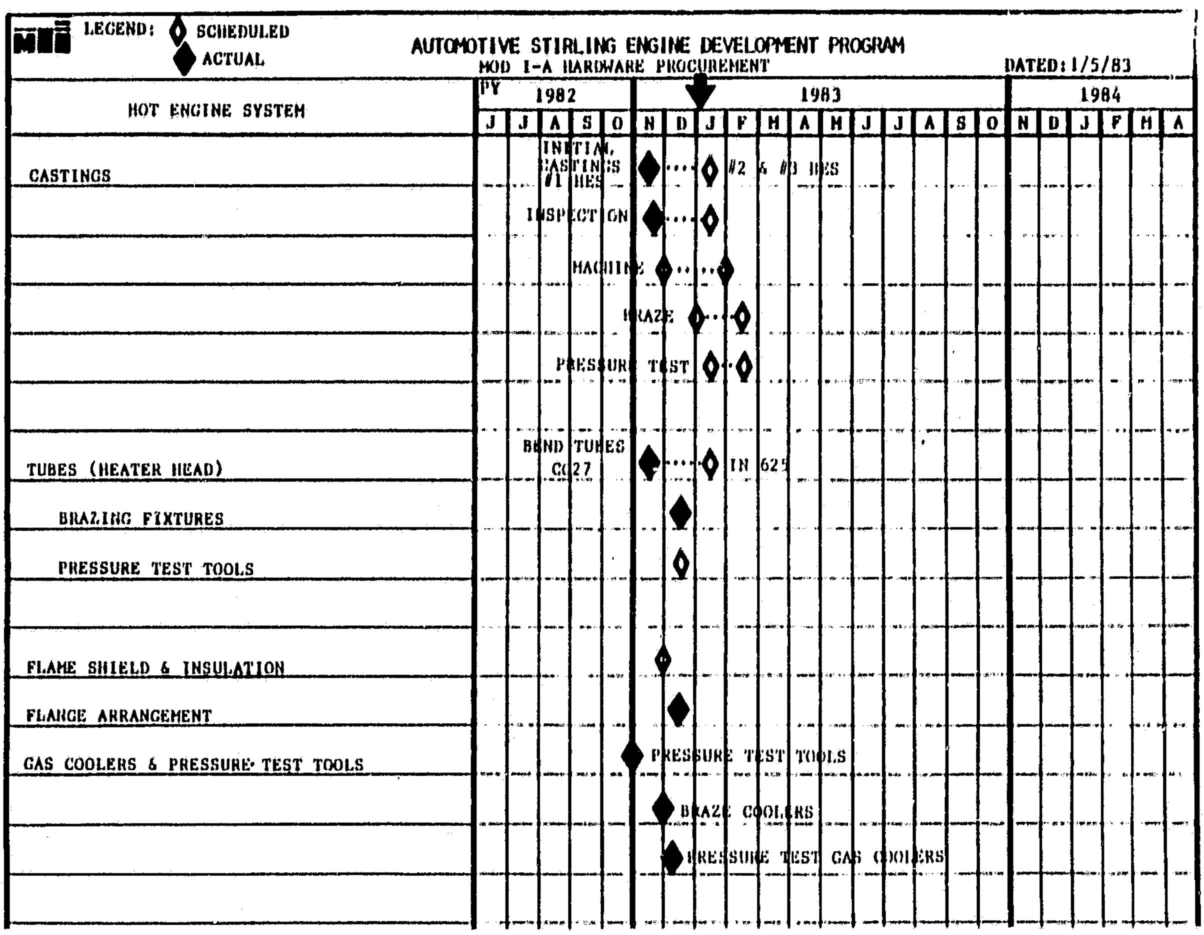

Figure 3-18 Mod I-A Hardware Procurement - Hot Engine System

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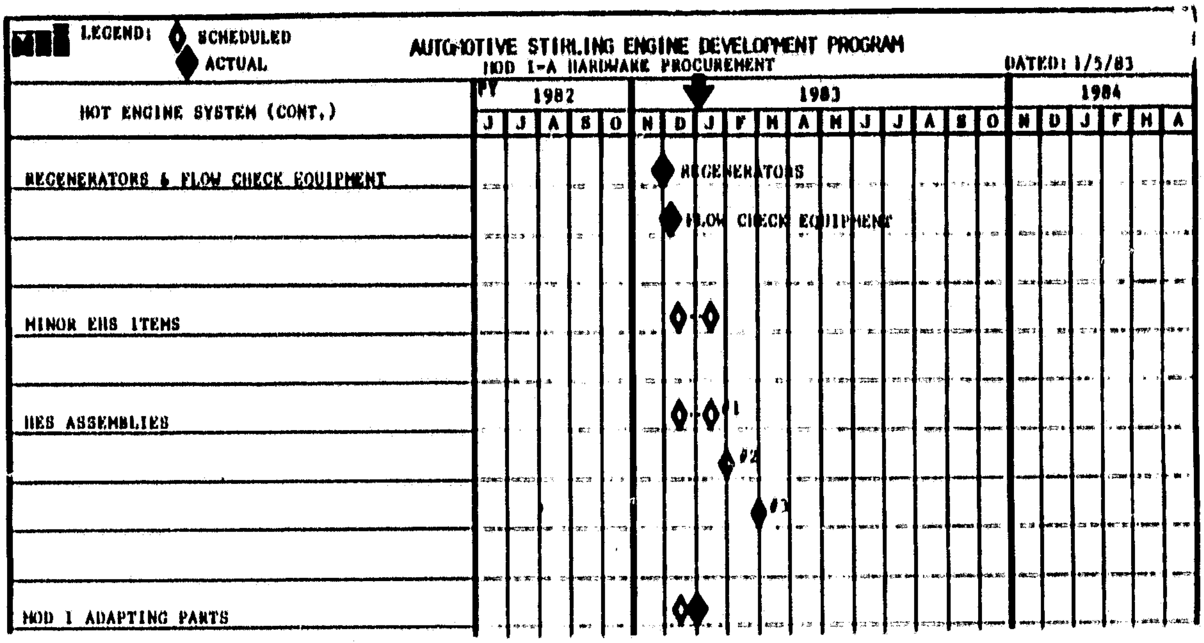

Figure 3-19 Mod I-A Hardware Procurement - Hot Engine System (Cont'd)

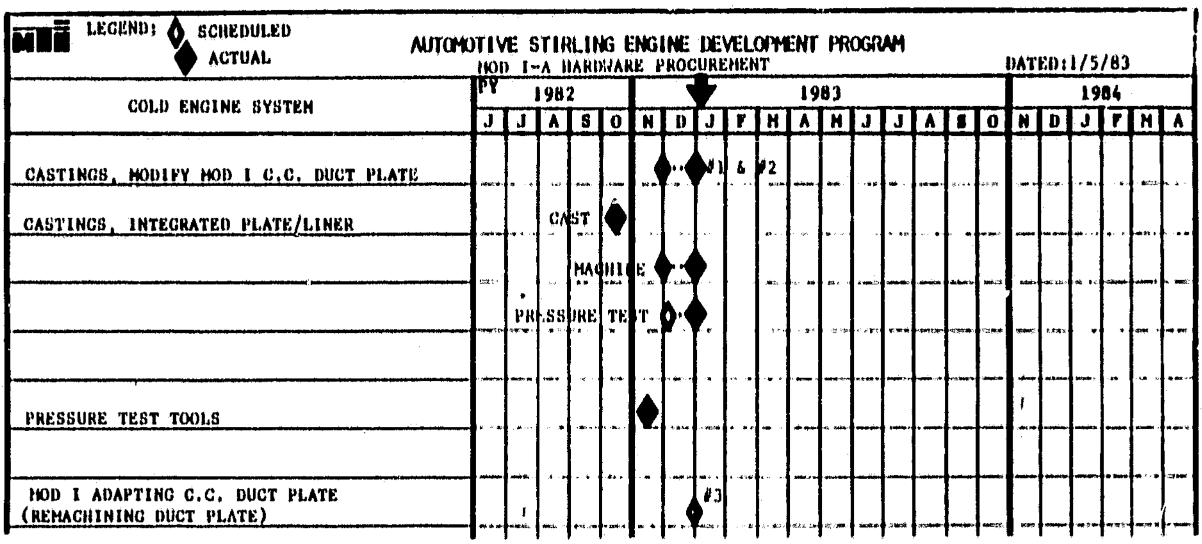

Figure 3-20 Mod I-A Hardware Procurement - Cold Engine System

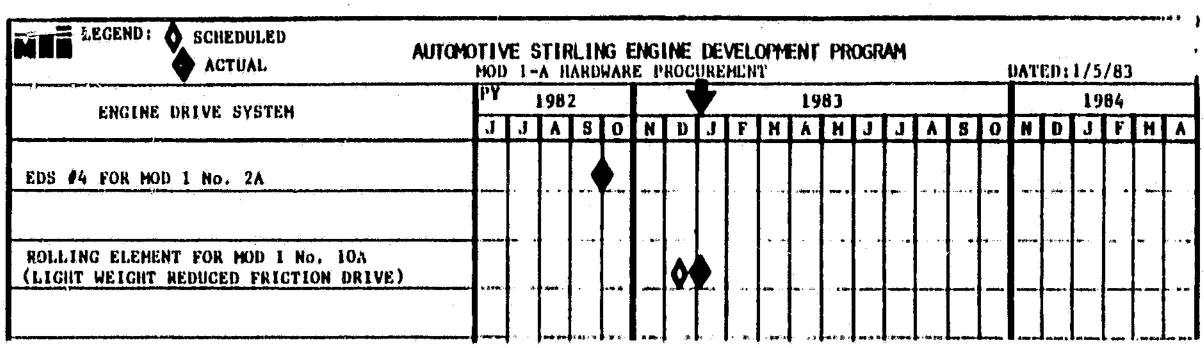

Figure 3-21 Mod I-A Hardware Procurement - Engine Drive System

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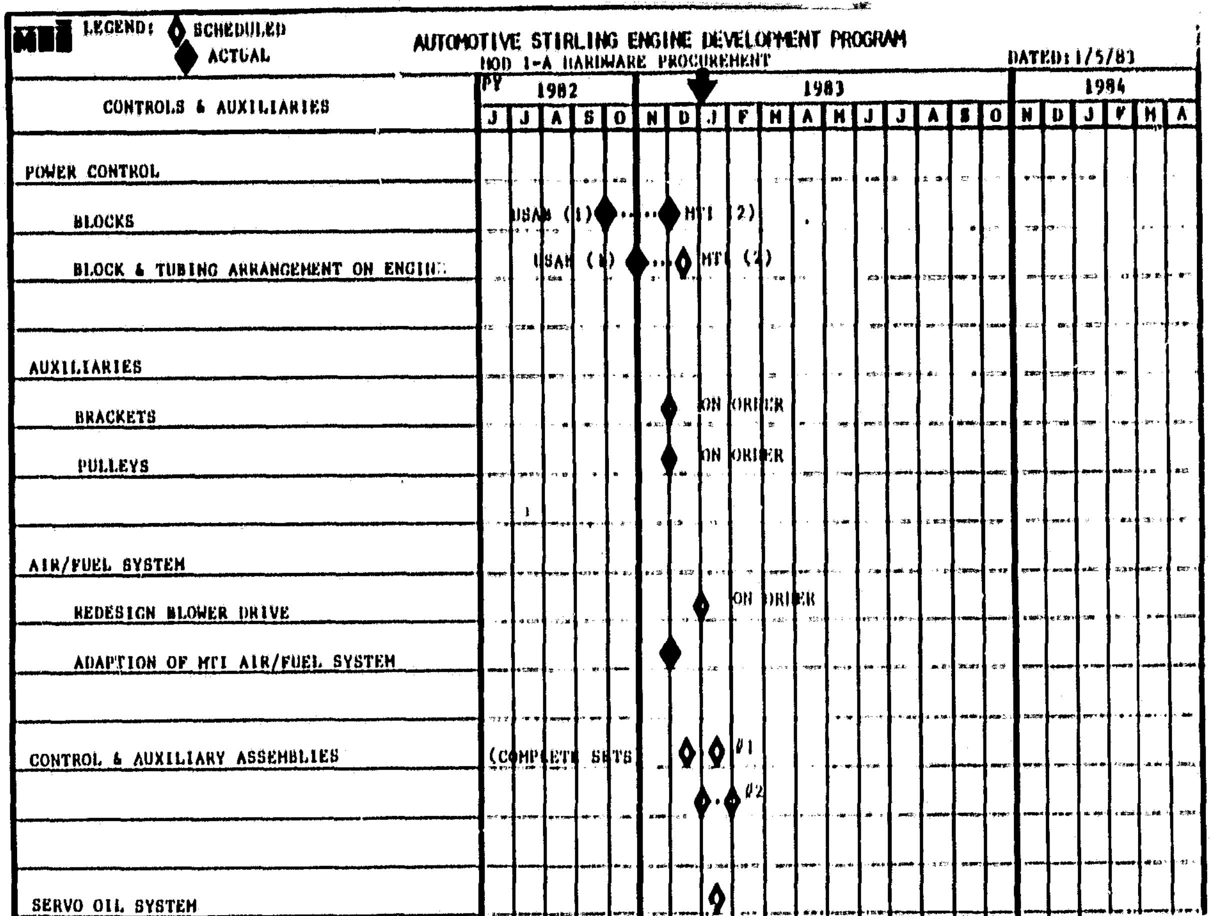


Figure 3-22 Mod I-A Hardware Procurement - Controls and Auxiliaries

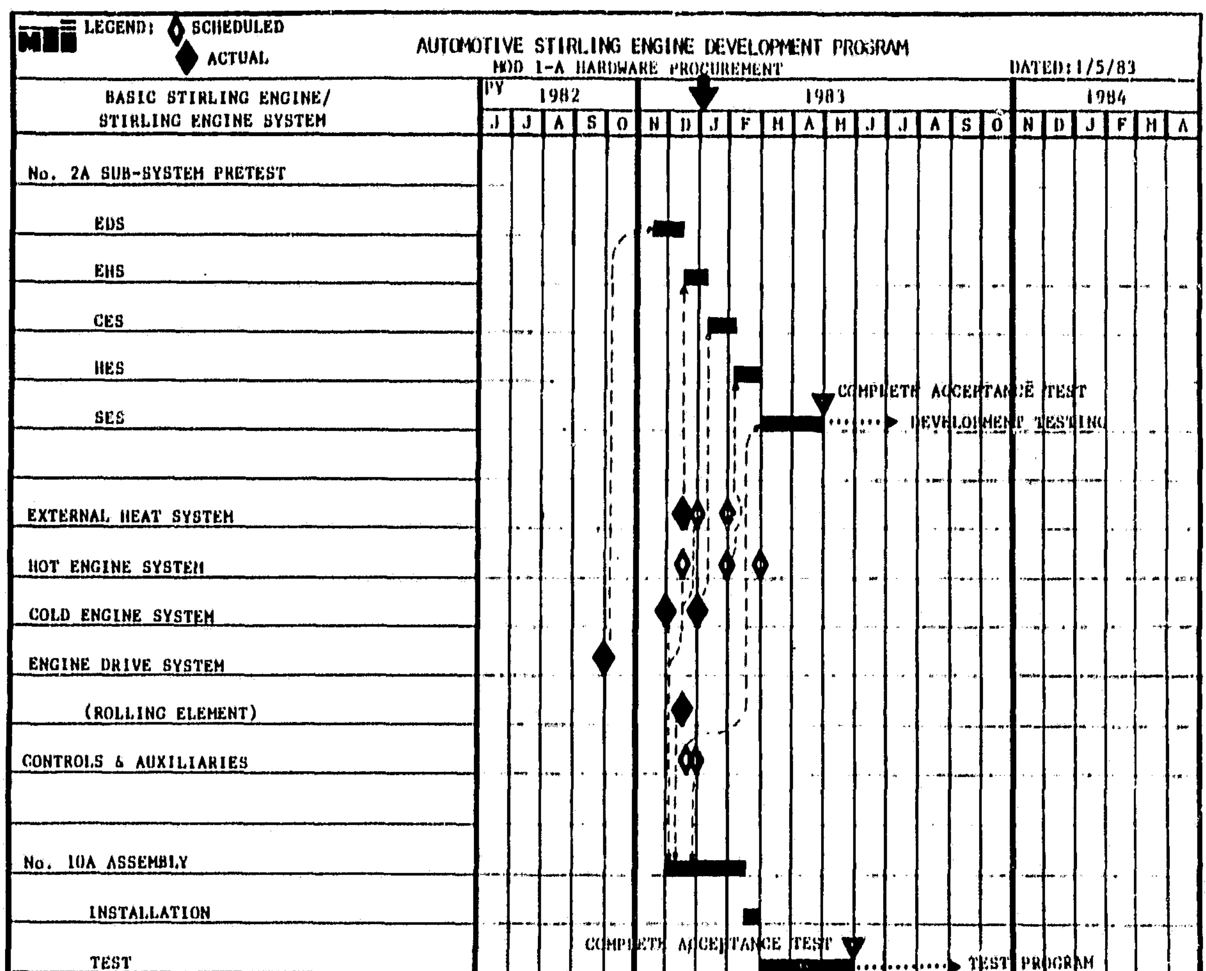


Figure 3-23 Mod I-A Hardware Procurement - Basic Stirling Engine/Stirling Engine System

RESD Design Update

The previous semiannual report (MTI Report No. 82ASE278SA2) reviewed concepts that were studied for a 45-kW (60-hp) automotive Stirling engine, and presented in some detail the U-4/V-4 hydrogen engines selected as prime RESD configurations.

During this semiannual report period, further refinements of the design were made, and documentation was initiated to produce a finalized engine design. Specifically, these changes included the following:

1) Simplified Power-Control System - The current mean pressure control system utilizes a linearly-operated valve to control hydrogen flow to and from the engine. The valve and actuator are relatively costly, and require a high-pressure oil system for servo activation. For the RESD, alternate control schemes are being configured to eliminate the current control valve. They include a rotary-distributor valve that, in conjunction with a rotary-control valve driven by an electric stepper motor, would eliminate the present control valve and several hydrogen system check valves. A direct pressure-control system utilizes a special simple, pressure-regulating valve in place of the power-control valve, and a modification of the rotary-control valve, which again replaces the power-control valve. All considered systems are less costly and simpler, which should provide improved engine operational reliability.

2) Extruded-Aluminum Gas Cooler - Further refinement of the gas cooler design has resulted in a design that utilizes a finned, aluminum cylindrical piece to replace the existing tubular cooler design. This concept (shown in Figure 4-1) is readily manufacturable, less expensive, and more reliable than the tube-type design.

3) Ceramic Preheater Design - This design is currently being utilized in rig tests. Preliminary development work on a ceramic heat-exchanger test section has indicated the feasibility of utilizing ceramic for this application. Consequently, the updated design uses ceramic in a polygonal-designed preheater with ten ceramic blocks that are rectangularly shaped for lower cost. Reduced maintenance and repair costs are attainable due to the capability of replacing a single block rather than the entire preheater should a localized problem arise. The latest V-4 version of the RESD is shown in Figure 4-2.

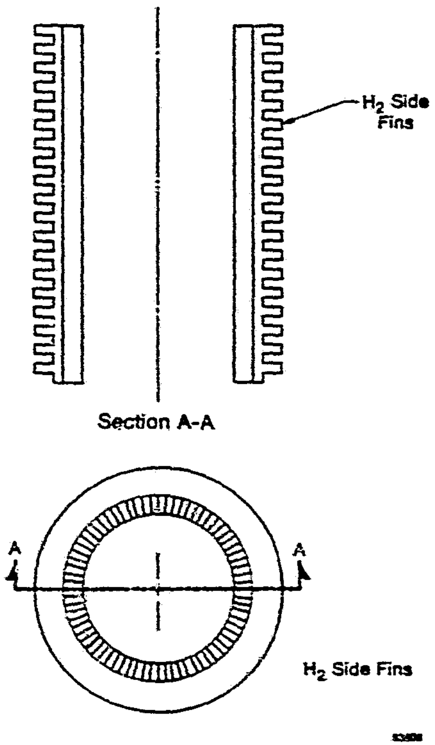


Figure 4-1 Extruded Aluminum Gas Cooler

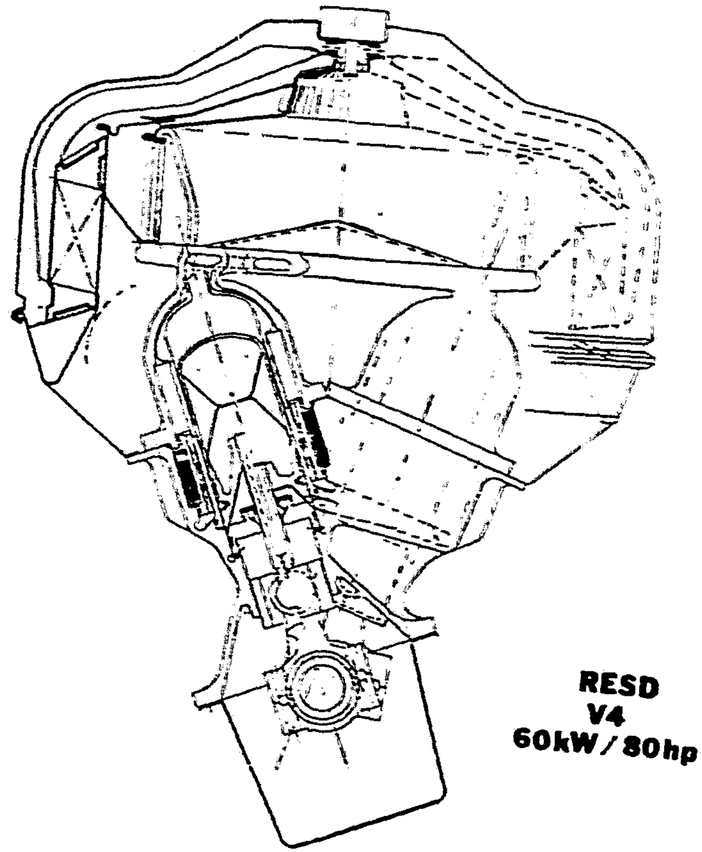


Figure 4-2 RESD V-4

V. ENGINE OPERATING HISTORY

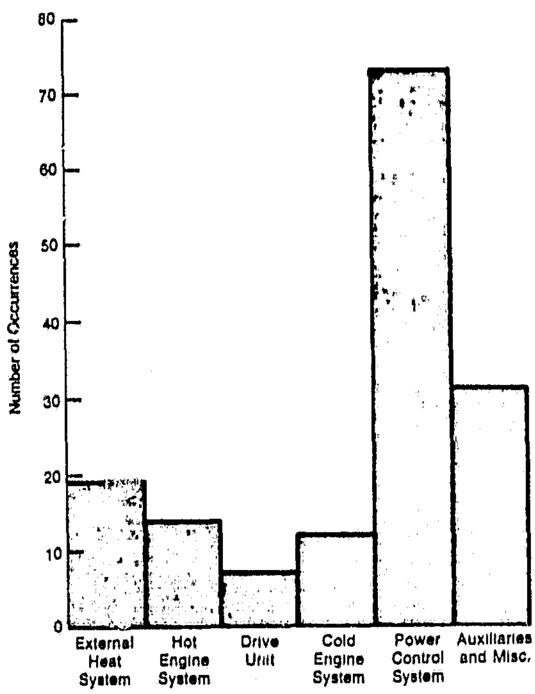
Quality Assurance Overview

Below is the status of the Quality Assurance Reports for this report period:

Total QAR's Issued	118
MTI QAR's Issued	92
USAB QAR's Issued	26
Total QAR's in Program to Date	366
Closeout Meeting 82-3 (10/5/82)	77
Open QAR's as of 12/31/82	68

An analysis of QAR's issued to date for Mod I engines showed the following trends:

Description	Occurrences
CGR Combustor	3
Igniter	5
Fuel Nozzle	6
Preheater	3
Heater Head	8
Flame Shield	4
Oil Pump	3
PL Seal	4
Valcor Dump Valve	5
MOOG Valve (Block #1)	17
Check Valves	6
Combustion Air Blower	6
Atomizing Air Compressor	4
Starter	3



Mod I Failures and Discrepancies Through December, 1982 Affecting Engine Operation

Table 5-1 is a summary of operating times and mean time between failures for all ASE Program engines as of December 31, 1982. The primary usage of each engine is:

- 40-4 - High-Temperature (820°C) Endurance Testing at USAB;
- 40-12 - Vehicle Demonstrations at MTI;
- Mod I No. 1 - Transient Test Bed at MTI;
- Mod I No. 2 - Dynamometer Testing at USAB;
- Mod I No. 3 - Dynamometer Testing at MTI; and,
- Mod I No. 10 - Dynamometer testing at MTI.

TABLE 5-1

SUMMARY OF MEAN OPERATING TIMES AND MEAN TIME BETWEEN FAILURES FOR ALL ASE PROGRAM ENGINES

ASE Engine	Operation Time *	Mean Operating Time Between Failures
40-4 (USAB)	8717.00	122.80
40-12 (MTI/Concord)	203.30	14.50
Mod I No. 1 (MTI)	644.20	64.40
Mod I No. 2 (USAB)	527.90	87.90
Mod I No. 3 (MTI)	299.90	299.90
Mod I No. 10 (MTI)	228.10	32.60

*Time prior to acceptance testing to present

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VI. FACILITIES

Below is a summary of the major facilities activities accomplished during the latter half of 1982:

- the Lister Engine regenerator durability testing is operational and currently in use;
- the Preheater Test Rig was built and activated (it is currently being used to evaluate ceramic perheaters);
- DAS hookup of the Combustion Performance Rig is complete; and,
- numerous smaller improvements to facilities have been made, as well as general upkeep of existing facilities.

APPENDIX - TERMS AND DEFINITIONS

Symbol/Term	Definition
Al	Aluminum
AMG	AM General
AOP	Average Operating Point
ASE	Automotive Stirling Engine
B	Boron
BOM	Bill-of-Materials
BSE	Basic Stirling Engine
C	carbon
°C	degrees Celcius
Cb	columbium
CES	Cold Engine System
CGR	combustion gas recirculation
Co	cobalt
CO	carbon monoxide
CO_2	carbon dioxide
Cr	chromium
CRU	Communications Register Unit
CVS	constant volume sample
DAS	Data Acquisition System
DC	direct current
DOE	Department of Energy
EDS	Engine Drive System
EGR	Exhaust Gas Recirculation
EHS	External Heat System
EHSTR	External Heat System Transient Response Code
°F	degrees Fahrenheit
Fe	iron
ft	foot
ft^2	square foot
ft^3	cubic foot
FY	fiscal year
g/mi	grams per mile
g/s	grams per second
HC	hydrocarbon
HES	Hot Engine Sytsem
hp	horsepower
HTP-40	High Temperature P-40 Engine
Hz	hertz
in	inch
kg	kilogram
ksi	thousand pounds per square inch
kW	kilowatt
lbs	pounds
LRFD	Lightweight Reduced Friction Drive
m	meter
mi	mile
mm	millimeter
Mn	manganese
Mo	molybdenum
Mod I	first-generation automotive Stirling engine
Mod I-A	first upgraded Mod I engine design
Mod I-B	second upgraded Mod I engine design
MTI	Mechanical Technology Incorporated
MPa	megapascals
MQ	material quote
N	nitrogen
NASA	National Aeronautics and Space Administration
Ni	nickel
NO_x	oxides of nitrogen
NTU	number of transfer units
PL Seal	Pumping Leningrader Seal
P_{max}, P_{mean}, P_{min}	working gas pressures
psi	pounds per square inch
psig	pounds per square inch gauge
PTFE	polytetraflouroethylene
P-V	pressure-volume
QAR	Quality Assurance Report
RESD	Reference Engine System Design
RFD	Reduced Friction Drive
rpm	revolutions per minute
s	second
SES	Stirling Engine System
Si	silicon
S/N	serial number
SS	Stainless Steel
Ta	tantalum
TTB	Transient Test Bed
USAB	United Stirling of Sweden
VE	Value Engineering
W	tungsten
λ	$\frac{\text{air/fuel}}{\text{air/fuel}_{\text{stoichiometric}}}$
η	efficiency
$\Delta\eta$	efficiency difference
ΔWt.	weight difference
ΔPower	power difference
α-I	alpha engine I
α-II	alpha engine II
ℓ/min	liters per minute